DESIGN AND ANALYSIS OF THE

INTAKE SYSTEM OF A FORMULA SAE CAR

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SUMMARY

This study takes a look at the design process of the air intake system of the FSAE car. Over the years, much of the design of this system had been carried out through an iterative trial and-error process, so the study attempts to identify the scientific and engineering principles pertaining to the design of this system. The intake system is being sub-divided into various components, and the relevant principles will be discussed. Following that, data is collected from the engine cylinders, cam-profile, intake valves etc. and a simulation model of the engine will be developed. This model is then being applied, sequentially, to the various components. Flow analysis for individual components are carried out, and verified against performance simulations of the entire engine system, followed by physical testing of several of the components using a flowbench

ACKNOWLEDGEMENTS

This thesis would not have been possible by my own effort alone, and I would therefore like to dedicate my appreciation to the following:

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Thank you.

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1. INTRODUCTION

This thesis is a study on the design of the various parts of the Air Intake system, specifically one that suits the requirements and restrictions of the Formula SAE Collegiate Series competition.

The air intake system is a major component required by the Engine system, and is critical in determining the amount of power produced by the engine system. However, it is made up of many sub-components that need to work well, and together, in order for the engine to perform up to par. With the introduction of a mandated air-restrictor, the off-the-shelf engine is unable to continue using the air intake system that was originally designed for it.

Over the years of the FSAE project, the air intake system had seen many iterations and improvements, allowing the engine to produce increasing amounts of power. However, there is always room for improvement, and this study aims to create a foundation of knowledge on which to build the next generation of air intake systems.

This foundation is to be built upon the basic physics theories that govern the air intake system, the “good practices” of the industry, discussions on the variation in design for various components and how they affect performance, as well as to establish the procedures of making use of available tools to improve the quality of the final design of the air intake system.

These tools, such as Computational Fluid Dynamics software, aid the design process through virtual simulations, including data acquisition and analysis of design variations for better information on the effects without cycling through the manufacturing and assembly processes.

Other tools, including test rigs, such as a Flow Bench, are used for physical testing of manufactured components of the air intake system, and aid the designer in the validation of his design, which, when compared against calculations and simulations results, provides the confidence that the system is able to perform its designated task, to the required levels of performance.

Besides the discussion on the tools that can be used, the report segments and discusses in parts, the various components of the air intake system, how to design them specifically for a particular engine and type of performance, and the considerations for the peculiarity of the engine that the FSAE team is using.

Another component of the entire design process is the physical testing of the completed air intake system on a dynamometer with a running engine, but will not be covered due to limitations in resources,

Otherwise, this report seeks to be a basic collection of information (complementing those that have been extensively studied, documented and are readily available) required to design and analyze an air intake system for a normally aspirated Formula SAE race car engine.

Organization of the Thesis

This report is generally organized into three main chapters, excluding the introduction and conclusion.

The first chapter looks into the existing knowledge on the intake systems, the design parameters and restrictions and some basic knowledge on the internal combustion engine and the air intake system on which this study was built upon.

The second chapter looks into the design of various components of the air intake system, sub-dividing them into various segments and using the applicable theories for each.

The third chapter then looks into various simulation and analysis methods, such as using flow simulation to analyze various components and configurations, as well as using a flow bench to test and verify manufactured components.

2. FUNDAMENTALS

This section highlights the basics of the engine system that the intake system is designed for. It specifies the scope around which this particular study of the intake system design is carried out.

INTERNAL COMBUSTION ENGINE

An internal combustion engine is one in which the engine has a combustion chamber in which a mixture of fuel and oxidizer is ignited to generate power. It is particularly characterized as an engine in which the working fluid is being ignited and expanded to gain mechanical energy that can be harnessed. This is opposed to external combustion engines in which the working fluid and the combustion elements are kept separated.

There are many different types of internal combustion engines available in the market, mostly of the reciprocating type, although there are others such as the rotary Wankel engines that are popular among today's automotive manufacturers. The engines used in the FSAE competition are primarily piston-type reciprocating engines, and while the competition is dominated by race cars sporting inline-4-cylinder engines, the team has been using a V-twin for the last three years, particularly for its advantage in power-to-weight ratio.

Among reciprocating engines, there is also a distinction between the four-stroke spark ignited engine we use, as compared to two-stroke engines or pressure-ignited engines such as the Diesel engine.

FOUR-STROKE CYCLE

The current dominant design of engines revolves around the four-stroke cycle, otherwise known as the Otto cycle. This particular design uses gasoline as the combustion fluid. Another four-stroke engine design, the diesel engine, will not be discussed here. The four-stroke gasoline engine had been the common choice due to a good combination between power, economy and environmental standards. These factors are achieved chiefly through the separation of each cycle, separating fresh air from spent and burnt air-fuel mixtures. The four-stroke cycle is so called because of the cyclic motion of the piston, within each cylinder, that travels through four strokes during each engine cycle. These four strokes are commonly known as the Intake, Compression, Power and Exhaust strokes and are shown in Figure 1

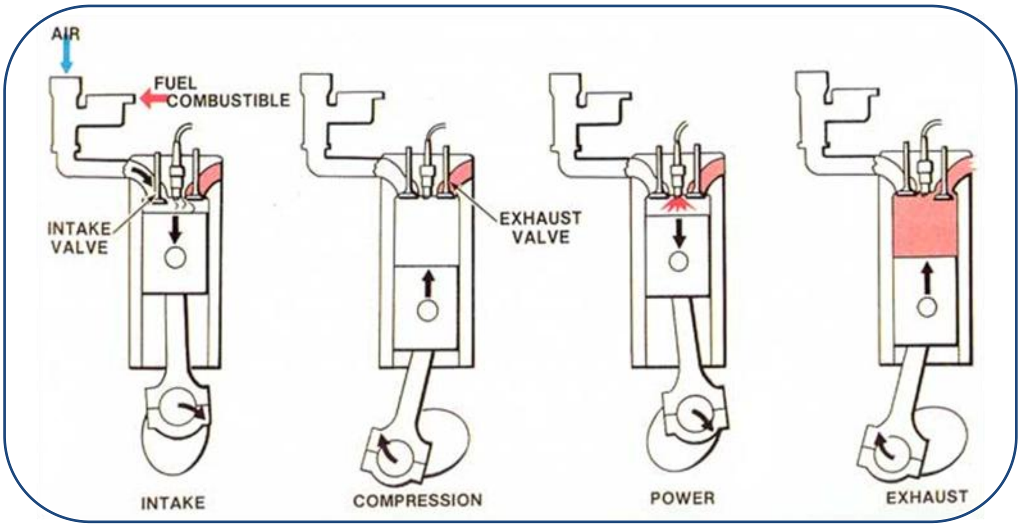


Figure 1: Four-stroke Cycle

Specifically, the Intake stroke has the piston moving from Top Dead Centre (TDC) to Bottom Dead Centre (BDC) with the intake valves open, effectively drawing air-fuel mixture into the cylinder. The

Compression stroke sees the closing of the intake valve, and the piston moving back up to TDC in an action that compresses the air-fuel mixture in the cylinder from atmospheric pressure to somewhere in the range of 200 psi. Near the end of the compression stroke, the spark ignites the compressed air-fuel mixture, raising the temperature of the gases in the cylinder to the range of few thousands of degrees, and increasing the pressure by up to 100 times. This increase in pressure then drives the piston downwards, in the Power stroke, which is when energy is transferred from the piston's movements to the crankshaft. The following Exhaust stroke is to evacuate the spent gases in the cylinder, through the now-open exhaust valve, by the piston's motion from BDC to TDC.

Apart from these four commonly known stages in the 4-stroke cycle, there are three other key states that are of significant note, but are less known. They are the periods of post-BDC intake, intake/exhaust overlap and early exhaust opening.

The first of these states refers to opening the intake valves after the intake stroke, during the compression stroke, even though the piston had reached the bottom most position at BDC. While it is counter-intuitive to open the intake valves while the overall cylinder volume is decreasing, it serves the purpose of allowing even more air to enter the cylinder. This is because air takes considerable time to be charged into the cylinder, flowing through the narrow openings of the intake valves, and flow rate that has been built up while the piston is moving downwards and drawing in air will continue to sustain, even though the piston has reached the lowest point. Opening the intake valves beyond the point of BDC, therefore, harnesses the inertia of the moving air to drive even more air into the cylinder, increasing the effective charge in the cylinder.

The second state happens near the end of the exhaust stroke, until the initial durations of the intake stroke. During this period of time, it would be observed that both the intake and exhaust valves are open, and while, again, it would seem counter-productive that fresh air-fuel mixture might be drawn into the cylinder through the intake valves, and dispelled immediately through the exhaust valves, the overlap was actually created to take advantage of the exiting exhaust gases to initiate the suction of fresh air into the cylinder. Subsequent to the combustion, the greatly expanded exhaust gas mixture would be quickly forced out of the exhaust valves. Achieving a certain volume flow during the exhaust stroke, the inertia of the exiting gas would leave behind a negative pressure void, in the cylinder, at the end of the exhaust stroke, and therefore opening the intake valves at this point of time will allow the negative pressure to be used to create a vacuum for the fresh air to fill, initiating the intake process and jump-starting the intake flow creation process. Once again, this would aid the engine in drawing more air quicker, again improving the volume of the air charge in the cylinder, improving volumetric efficiency and eventually power output.

The third state, in which the exhaust port opens earlier, is for the purpose of allowing more air by making use of some of the energy of the power stroke to expel exhaust gas. While it consumes some power from its own power stroke, it actually reduces the load on the other cylinders compared to if the exhaust gas is purely pumped out of the cylinder by the upwards stroke of the piston.

ROLE OF AIRFLOW

The entire engine system can be likened to a massive air-pump, or the respiratory system in a human being. Air from the environment is drawn in, does its work, and is expelled. In a human body, work is done when oxygen is being drawn into the lungs. In an engine, this same oxygen is used to support the combustion of fuel, gasoline in this particular study, and this combustion in turn generates energy that drives the output of the engine, through the transmission system, and to the wheels that propel the race car.

The entire system involving obtaining air from the environment, using it, and expelled consists of the air intake system, the engine’s cylinders, and the exhaust system. The better flow of air through this systems will lead to a better output from the engine. However, there are a lot of inefficiencies that might be introduced by a poorly designed air intake system, which eventually robs the engine of its power.

It is therefore critical for airflow to be maximized by improving and fine-tuning the air intake system,

VOLUMETRIC EFFICIENCY

Volumetric Efficiency is a measure of the Cylinder Charge. It defines the amount of fresh charge that can be sucked into the cylinder, as a ratio of the theoretical mass of air that can be contained in the cylinder. Essentially, for a normally aspirated gasoline engine, it is the ratio of the trapped volume of gas to the volume of the cylinder.

Volumetric Efficiency, $\lambda_a = \frac{m_g}{m_{th}} = \frac{m_g}{V_H \cdot \rho_{th}} = \frac{V_G}{V_H}$

$m_g = V_g \cdot \rho_g$ *= mass of air that is trapped in cylinder*

m_{th} *= theoretical mass of air that can be trapped in cylinder*

$V_H = V_{th}$ *= Volume of Cylinder, which equals theoretical Volume*

V_g *= Volume of gas trapped in cylinder*

Assumptions:

Theoretical density of air in cylinder, ρ_{th} *= Density of ambient air,* ρ_g

The Cylinder Charge of an engine determines the amount of power it can produce, and varies with air-fuel ratios as well as RPM. At the exact stoichiometric ratio, the power produced would be proportional to the mass of air (or air-fuel mixture) being supplied to the engine, otherwise known as the “charge” of the engine.

In the scenario where the only method of charging the engine is by the vacuum pressure created by the expanding of the combustion chamber when the piston falls, it is able to achieve a maximum volumetric efficiency of 100%. This figure, however, would be reduced by the efficiencies of the air intake system, such as restricted flow through the throttle body and intake valves, the energy loss through friction with the inner walls of the air intake system, as well as the propagation of reduced air pressure when the cylinder’s vacuum initiates.

In order to squeeze out more power from the engine, more air has to be forced into the combustion chamber, leading to development of forced induction systems, such as super-chargers and turbo-chargers, the former being a mechanically (through belts or gears attached to the rotating engine shafts) or electrically driven air compressor, while the latter, also known as a turbo-super-charger, is an air compressor driven by the expelled exhaust gases, through a turbine.

While the topic of forced induction is not a part of this report, methods of improving volumetric efficiency through other means will be discussed in the following chapters, for example using tuned runner lengths to create acoustic waves of increased pressure during the intake valve's opening, to allow more air into the cylinder.

COMPRESSION RATIO

By definition, the compression ratio of an engine is determined by the ratio of the largest volume of the cylinder (typically at the Bottom-Dead-Centre position of a piston engine), to the smallest volume of the cylinder (at the Top-Dead-Centre of the stroke). This defines how much a volume of air, assuming that the volumetric efficiency is 1, is being compressed by the piston.

The typical value for a high performance engine, particularly one which is used on a race car or motorcycle, will fall in the range of 13:1, indicating that the volume of air in the cylinder is being compressed 13 times, at the maximum. The SXV550 engine runs slightly lower at a compression ratio of 12.5:1.

In comparison, the typical trend of compression ratios would suggest that a higher compression ratio encourages detonation, a phenomenon in which the suddenly pressurized (and therefore self-heated) air-fuel mixture in the cylinder will ignite by itself, without the ignition spark. This causes two possible scenarios, the first of which being an excessively advanced ignition, in which the detonation of the air-fuel mixture in the cylinder creates a pressure increase that works against the upwards compression motion of the piston, causing stress on the crankshaft or con-rods. The audible result of this is known as “engine knocking”, in which a sound resembling the shaking of a bag of marbles will accompany the usual ignition sounds. The prolonged experience of knocking leads to damage in the con-rods, crankshafts and eventually engine seizure.

The second possible scenario is when the compressed air-fuel mixture initiates a flame front at one portion of the cylinder, while the spark ignition initiates another flame front from its location. These two flame fronts will meet and cancel each other, suggesting that there will not be a single, large explosion in the chamber, but instead multiple, smaller explosions, generating less gas expansion and consequently less power being transmitted to the crankshaft.

On the other hand, a lower compression ratio leads to poorer performance in terms of both power output and therefore fuel economy. The amount of emissions, conversely, will be higher. These three effects are largely attributed to the inefficiencies of igniting an improperly compressed air-fuel mixture, resulting in an inability to completely combust the air-fuel mixture.

CHOKED FLOW

The phenomenon of a choked flow system is one pertaining to compressible flow, such as that of air in the atmospheric environment flowing through the air intake system, and into the engine's cylinders. In the FSAE context, the main location in which choked flow is likely to develop would be at the air intake restrictor. It is formed when air flows across a path with a decreasing cross-sectional area.

The mass flow rate under a choked flow condition would be defined by the following formula:

Mass Flow Rate, $\dot{m} = C\ A\ \sqrt{k\ \rho\ P\ (\frac{2}{k+1})^{(k+1)/(k-1)}}$*; kg / s*
C = discharge coefficient
A = discharge hole cross-sectional area, m^2
k = c_p/c_v *of the gas*
c_p *= specific heat of the gas at constant pressure*
c_v *= specific heat of the gas at constant volume*
ρ *= real gas density at P and T,* kg/m^3
P = absolute upstream pressure of gas, Pa

In order to complete the calculations, the various values of the air's property need to be plugged into the formula. Certain assumptions are made of the condition of the air, such as its composition, as air is a mixture of various types of gases. The exact value would have slight differences but should generally be in the same range of values. The calculation below shows the values used:

C = 1
A = 3.142 x $10^{-4}\ m^2$ *(20mm diameter restrictor)*
c_p *= 29.19 J / mol. K*
c_v *= 20.85 J / mol K*
k = 1.4
ρ *= 1.2041 kg /*m^3
P = 101325 Pa
Mass Flow Rate, $\dot{m} = C\ A\ \sqrt{k\ \rho\ P\ (\frac{2}{k+1})^{(k+1)/(k-1)}}$ *= 0.075 kg / s*
Volume Flow Rate = 0.0624 m^3 */ s*

In this calculation, a discharge coefficient of 1 is taken, though the number is typically smaller than this. The discharge coefficient is the ratio of the actual flow rate to the ideal flow rate of the gas, given the same initial (before restriction) conditions. The calculations therefore provide the ideal mass flow rate, and the result taken as a reference maximum.

ENGINE CONFIGURATION ON A FSAE CAR

Type of Engine

On a Formula-SAE race car, the rules restrict the usage of engine types to being

i. a reciprocating piston-type engine;
ii. utilizing a four-stroke combustion cycle;
iii. of a displacement not exceeding 610cc.

In addition to the restrictions on the engine, it is required that an engine operating on gasoline as a fuel type, must have an air intake restrictor through which all of the air entering the engine must pass. On the intake manifold system, the restrictor has to be placed between the throttle mechanism and the engine itself. It must also not be movable or flexible in any way. This restrictor has to be circular in shape, and limited to 20mm in diameter.

Limitations to air intake dimensions are introduced as a bid to limit the overall power of the engine, and subsequently the vehicle, so as to reduce the speeds of the built vehicles on the track. It also adds an element of design variation as an off-the-shelf engine cannot be directly used on the car, spurring students to have to design a suitable air intake system, to reduce the impact of the air restrictor on the entire engine system.

Location of Engine

The dominant design of the Formula SAE race car has the engine located in the rear of the vehicle, driving non-steerable rear wheels. The location of the engine reduces any lengthy drive components by keeping close to the driven wheels. Apart from that, slight adjustments of the engine location in any of the three axes are determined by the weight distribution of the car, packaging, and a compromise to contain other components within a given envelope of the car.

Engine Selection and Comparison

Since the inception of the Formula SAE project, the team has gone through several iterations of choices of engines. The two engines that were used predominantly were from the Honda CBR600F4i from 2004 to 2009 and the Aprilia SXV550 since 2010. A comparison of the two engines is shown below in Figure 2:

CBR 600 F4i

- **Configuration -**
  - Inline 4-cylinder
  - DOHC
  - 599 cc
- **Weight -**
  - 58.2 kg
- **Horsepower -**
  - 109HP at 12500 RPM
- **Lubrication -**
  - Wet Sump

SXV 550

- **Configuration -**
  - 77° V - twin (2-cyl)
  - SOHC
  - 549 cc
- **Weight -**
  - 32.7 kg
- **Horsepower -**
  - 65.9HP at 10900RPM
- **Lubrication -**
  - Dry Sump

Figure 2: Comparison of F4i and SXV550 Engines

From the specifications shown in Figure 2 below, the apparent reasons for using the SXV550 becomes clear, with a higher power-to-weight ratio, the SXV550 engine would be a preferred choice of engine. In addition, it is about 25kg lighter, and with two cylinders instead of four, as well as an integrated dry sump for lubrication, it would provide a weight saving of over 30kg in all. Yet another factor pointing towards adopting the SXV550 engine, is that the FSAE race car has been operating mostly in the traction-limited region, despite using the largest available tire size as well as the more suitable tire compound, rather than the power-limited region of operation. This means that for most ranges of engine speeds, the FSAE race car is producing more power than it can transmit to the ground, suggesting that there is a tolerance for a reduction in the absolute value of output power.

The choice to move into using the SXV 550 engine was not without its problems, as it is an engine that is characteristically difficult to start. Its Engine Management System also reads and characterizes the engine timing through a single Crank Pulse Sensor, rather than the usual configuration of an additional Cam Pulse Sensor. An after-market Cam Pulse Sensor, integrated to a machined cam cover, was used to provide the third-party Engine Management System that the team uses with the required timing signals to operate.

In addition, other associated problems were a higher engine noise volume that could affect the mandatory sound test, exhaust ports on either ends of the engine that made it more difficult to package the exhaust system, and an odd-firing ignition pattern (cylinders fire at 283° and 437° crank angle intervals), as opposed to even-firing (uniformly at 360° crank angle intervals). However, there are work-around solutions for these problems, which allow the team to take advantage of the pros of this engine, but they are outside of the scope of this report.

Components of the Intake System

The intake system is made up of a few components placed in series, and lie between the atmospheric air and the intake valves of the engine. The primary function of the air intake system is to provide air to the engine. The key restriction placed upon the air intake system, particularly by the Formula SAE rules, is that all the air entering the engine will have to pass through a restrictor of less than 20mm in diameter. Other restrictions include the positioning of the throttle body, having to be before the restrictor, and that any equipment utilized in the attempt to achieve forced induction of air into the engine, such as a turbo-charger or a super-charger has to be placed after the restrictor.

In the following diagram (Figure 3) of the air intake system of 2011, the individual components will be highlighted. The subsequent chapter will elaborate on each of the individual components.

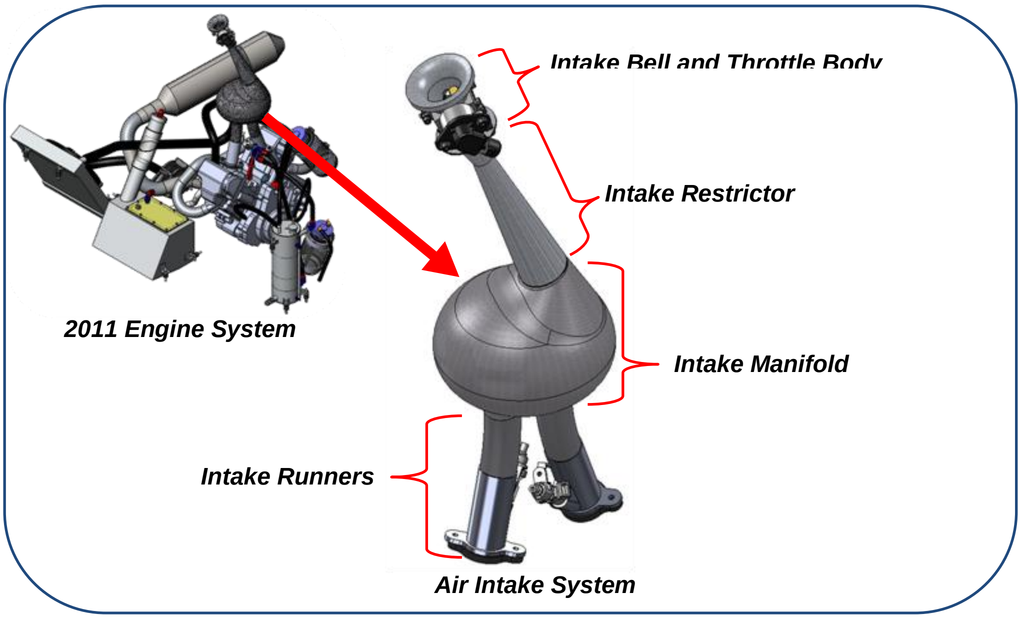


Figure 3: Components of the Air Intake System

3. INTAKE COMPONENTS

CYLINDERS – BLOCK AND HEAD

The cylinder can be considered as the final destination for airflow in the intake cycle. Having passed through the entire intake system, the mixture of air and fuel will be inducted into the cylinder, through the intake valves, in the Intake stroke. The subsequent Compression stroke will increase the pressure, with the ignition coils starting the combustion process near to the maximum pressure of the air-fuel mixture.

In this section, the design of the Aprilia SXV550's engine is being scrutinized, including the collection of data for various components of the cylinders, and creating a working model for the engine, to simulate the four cycles of the engine for further computer simulations and analysis of the intake system

Valve Lifts

The SXV550 engine is designed with a single-overhead-camshaft (SOHC), which indicates that both the intake and exhaust valves are being operated by a single rotating actuator. Due to the size of the cylinder bore, the intake and exhaust valves are placed significantly far apart, prompting a need for a rocker design to open one set of the valves. In this particular engine, the rocker is used to actuate the exhaust valves.

Also, the cylinder has a total of four valves, two for the intake, and two for the exhaust. The two intake valves are significantly bigger, for the reason that the oxygen-bearing air needs to enter the cylinder faster, as compared to removing the exhaust gases. In the exhaust stroke, the exhaust gases in the cylinder, pressurized after the combustion process, creates a large pressure gradient across the exhaust valves, allowing a faster flow. Comparatively, the pressure difference across the intake valves is lower, between the atmospheric pressure in the manifold, and the suction pressure of the falling cylinder. To compensate, the intake valve has to be larger, helping in improving the volumetric efficiency of the engine.

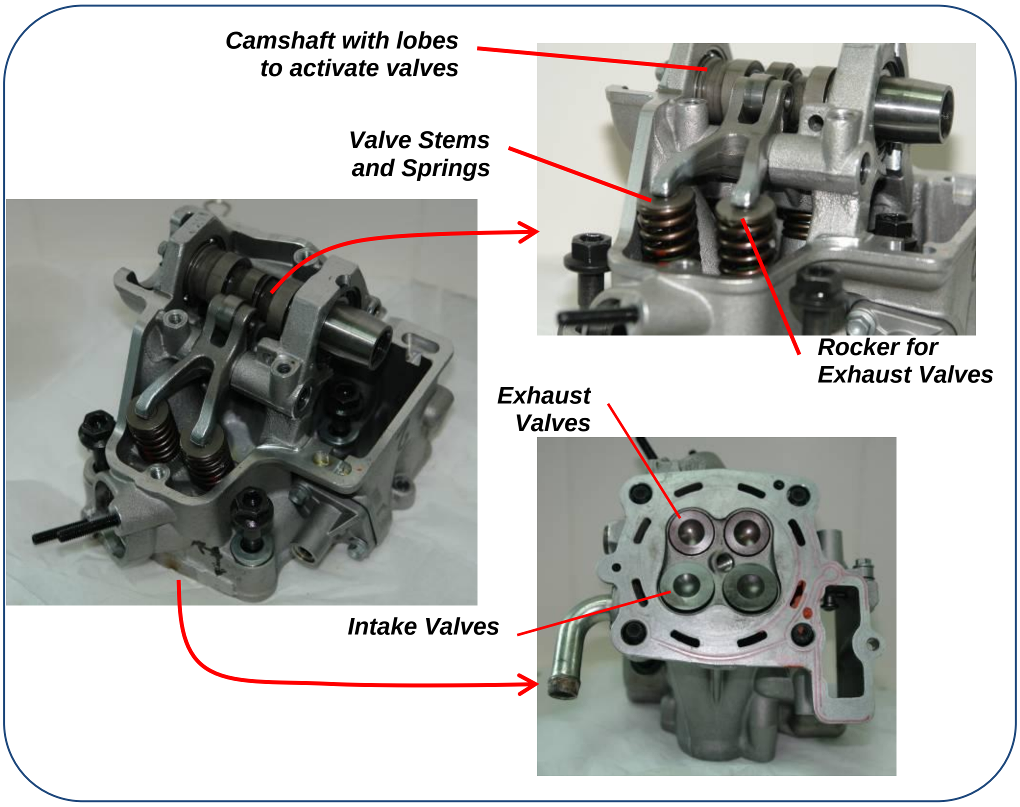


Figure 4: Cylinder Head

Figure 4 above shows one of the two cylinder heads of the engine, illustrating the 4-valve system used, and the SOHC that pushes against the valve stems to open the valves, which are returned by the compressed valve springs.

One critical factor that affects the design of the intake system is the amount of valve lift that the camshaft produces. The valve-lift data is usually a closely guarded information by each engine's manufacturer, however, a simple jig can be built to collect the information on the valve lift. Figure 5 below shows the jig that was designed for collecting information on the valve lifts.

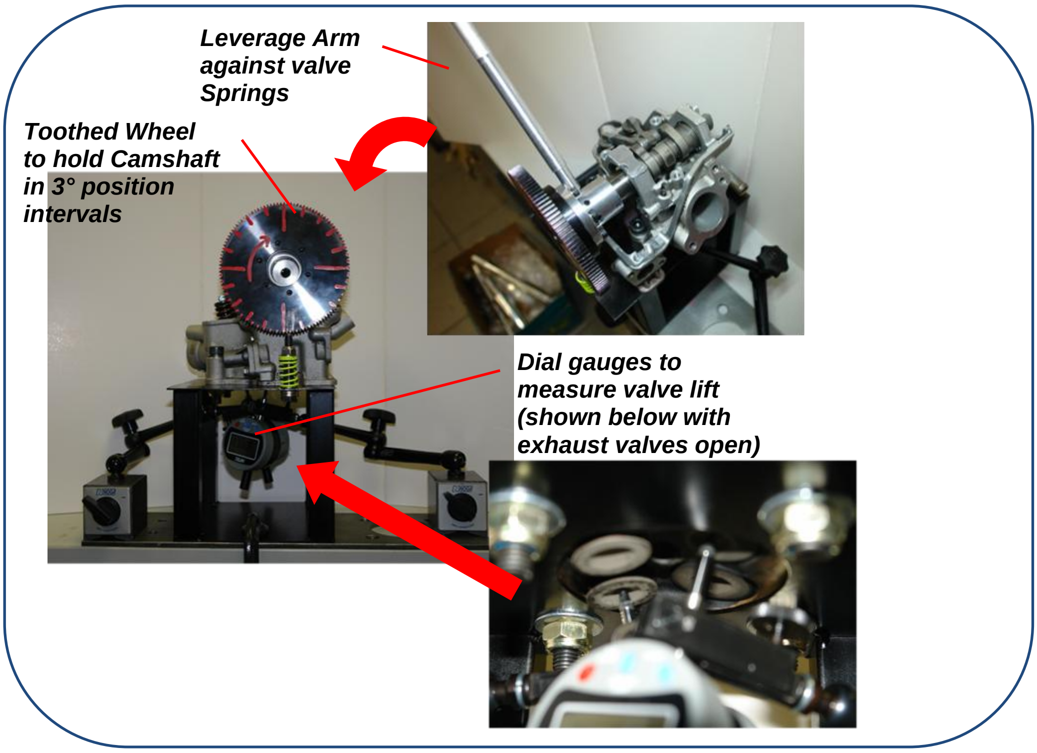


Figure 5: Measurement of valve lifts

Dial gauges are used to measure the minute movements of the valves, while a 120-teeth gear is being rotated and stopped at each gear tooth position. This gives a collected data reading of 3° intervals. The collected data was tabulated into a valve-lift chart, which was based on the 360° cycle of the cam-shaft. This was then referenced to the 720°

rotation of the crank, as it is a 4-stroke engine. The values were then graphed as shown below, with each 180° representing, in order, the Power, Exhaust, Intake and Compression strokes.

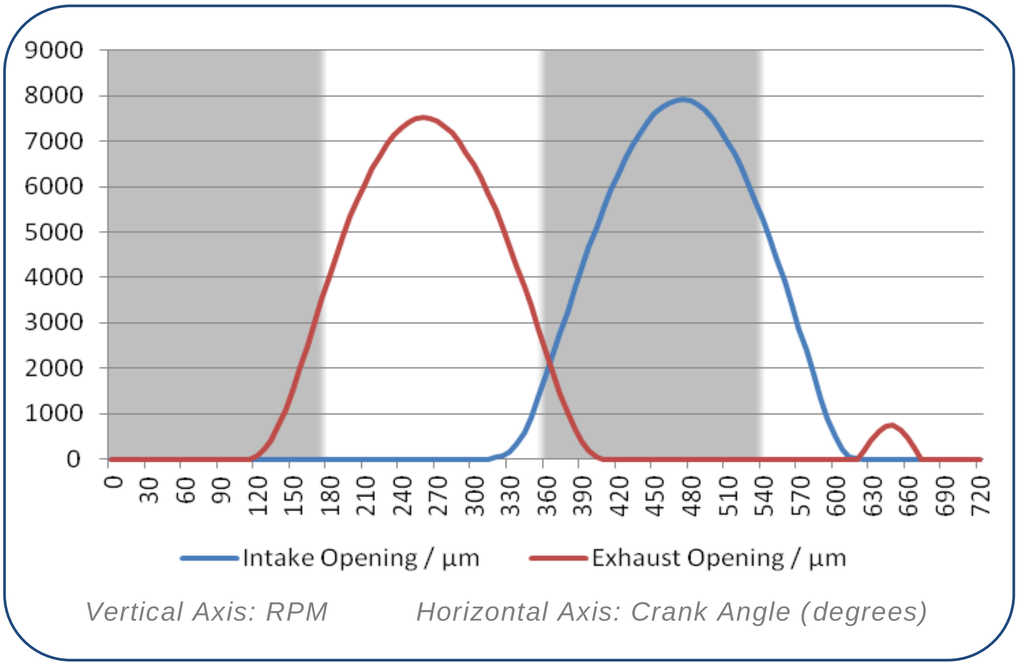


Figure 6: Intake and Exhaust Valve Lifts

As a derivation of the valve lifts, the cross-sectional area, created by the movement of the intake and exhaust valves, through which air flows in and out of the cylinder respectively, is being computed. This computation takes into account of the 45° sloped surface which is the sealing face between the back of each valve stem and the valve walls. Figure 7, illustrating the calculated area is as follows:

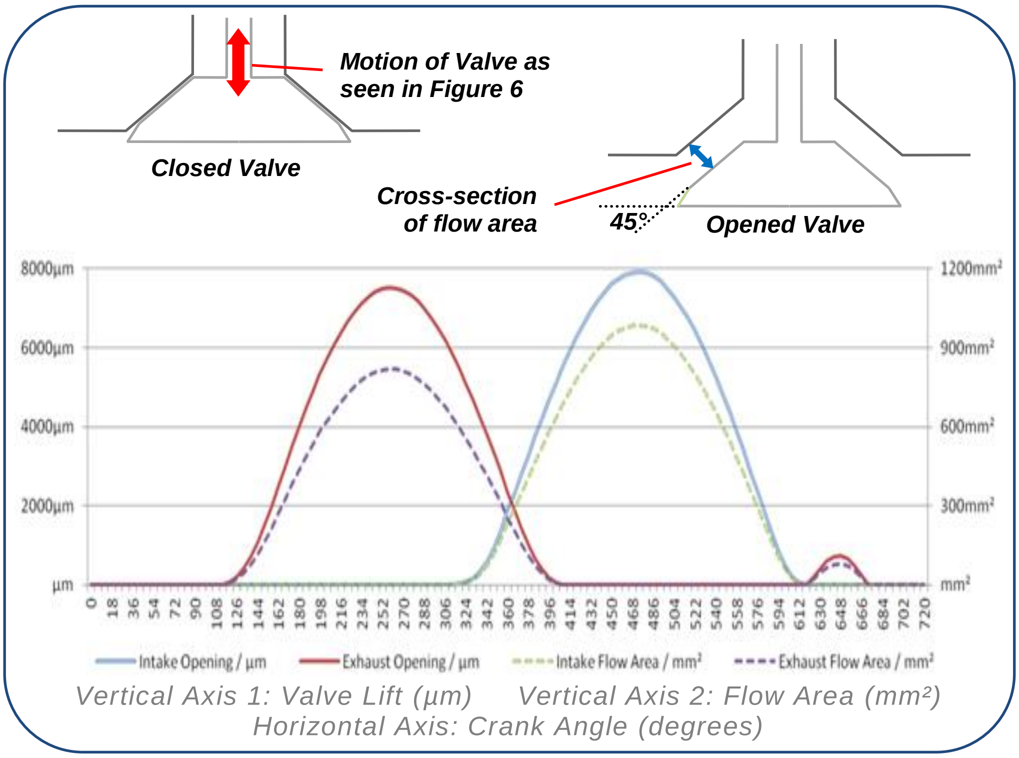


Figure 7: Flow-area created by opening valves

Cylinder Motion

A key value to be considered for the intake is the piston's speed. Essentially, it is the speed at which the face of the piston moves up and down within the cylinders' sleeves. This speed is affected by the RPM of the engine, particularly the speed of the crankshaft.

From the information provided by the manufacturers of the engine, the Stroke of each of the two cylinders of the SXV550 engine is 55mm (Refer to Appendix A). This implies that the face of the piston moves over a distance of 55mm during each stroke. In order to find the Piston Speed, the following calculations are used:

Revolutions per Second = RPM / 60
Time required for each cycle, t_{cycle} *= 60 / RPM s*
Mean speed of piston, $\overline{v_p}$ *= 55mm * 2 /* t_{cycle} *= (11 x RPM) / 6 mm/s*
Example – @ 3000 RPM, $\overline{v_p}$ *= 5500mm/s*
@ 8313 RPM, $\overline{v_p}$ *= 15241 mm/s*
= 914.4 m/min
= 3000 ft/min (noise test value)

As seen from the equations above, the Piston Speed is a factor of the RPM at which the engine is operating. Also, the calculated speed is a mean value, using the absolute distances travelled over the time of a cycle. In order to get a clearer picture of the speed cycle of the piston head, the actuator of the piston is being scrutinized.

Similar to many of the modern day engines, the pistons are driven by a crankshaft, which turns at the engine RPM. A connecting rod, or con-rod connects the crankshaft, at an offset from the central axis, to the under-side of the piston head, translating the cyclic rotational motion of the crankshaft into the reciprocating linear motion of the piston. Due to this motion translation, the speed of the piston head, assuming that the crankshaft is rotating at a constant speed, will then obey a sinusoidal pattern.

From this assumption, the maximum speed of the piston head is calculated as:

Maximum speed of piston, $v_{p,max}$ $= \overline{v_p}$ *x* $\sqrt{2}$
= $(\sqrt{2}$ *x 11 x RPM) / (6) mm/s*

The minimum speed of the piston would be at the maximum and minimum points, or TDC and BDC of each cycle. It then follows to assume that the maximum speed of the piston would be when the piston is halfway through its stroke, which would likely also coincide with the point where the offset shaft of the crankshaft, which connects to the con-rod, is perpendicular to the motion of cylinder, giving the largest moment arm. It is also initially assumed that the linear, vertical motion of a cylinder will adopt a sinusoidal motion when driven by the circular motions of the crankshaft.

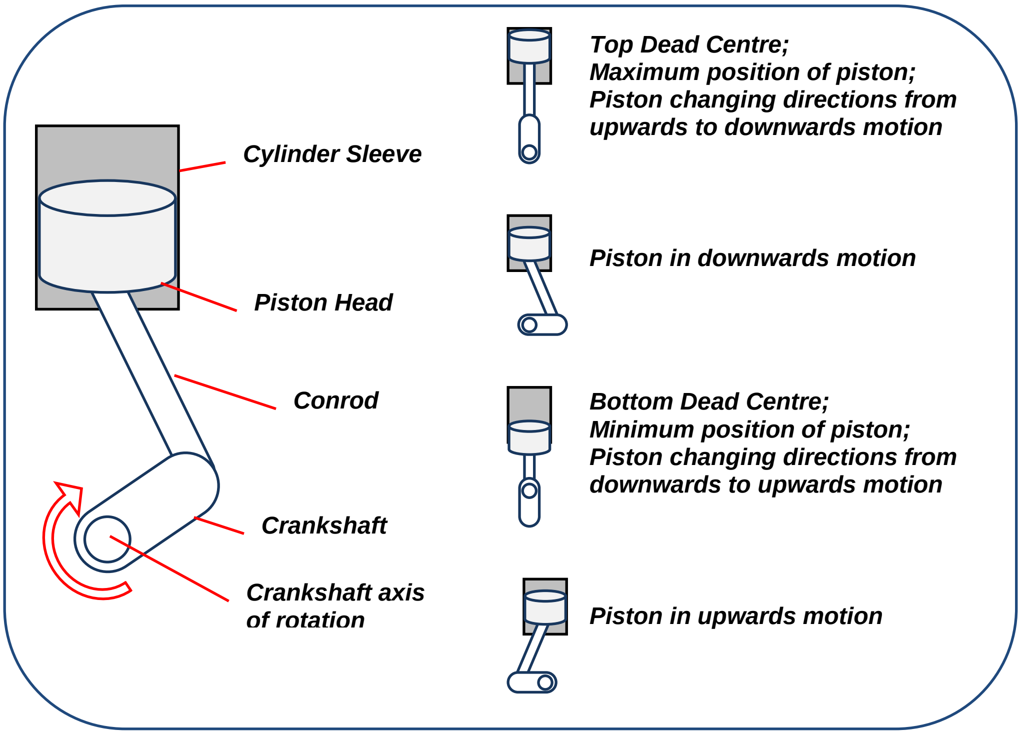


Figure 8: Cycle of piston motions

Consequently, with this combination of moment arms (the crankshaft and con-rod), a trigonometric equation can be formulated (as shown below in Figure 9) to more precisely calculate the position of the piston. In addition, the equation will allow for multiple differentiation processes to be applied, with respect to the crank angle, such that the velocity and acceleration can be obtained, albeit expressed as a function of the crank angle.

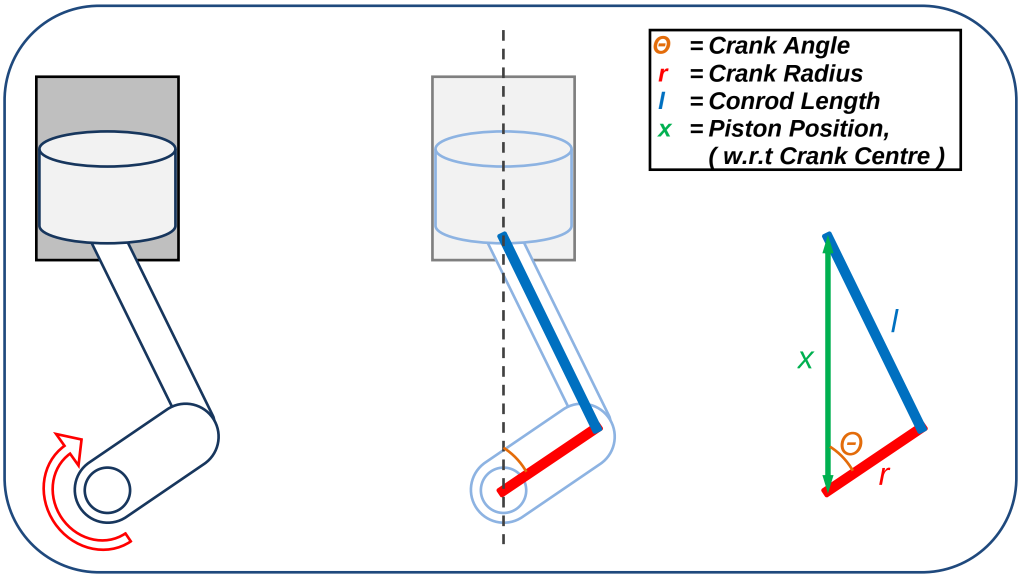
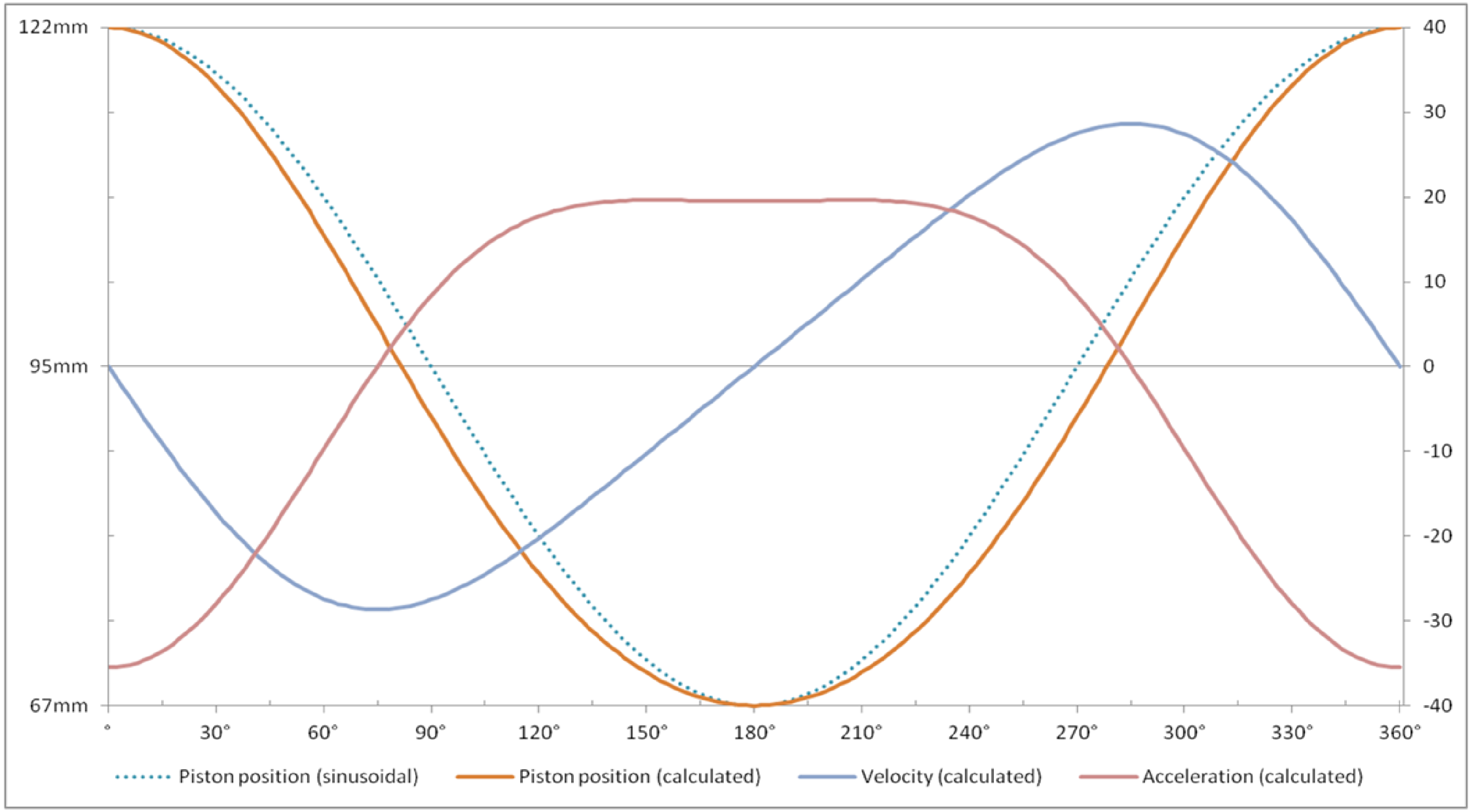


Figure 9: Trigonometric Expression of Piston Motion

Position of Piston, x	$= r\cos\theta + \sqrt{l^2 - r^2\sin^2\theta}$
Velocity of Piston, x'	$= {dx}/{d\theta}$
	$= -r\sin\theta - \frac{r^2\sin\theta\cos\theta}{\sqrt{l^2-r^2\sin^2\theta}}$
Acceleration of Piston, x''	$= {dx'}/{d\theta}$
	$= -r\cos\theta - \frac{r^2(\cos^2\theta-\sin^2\theta)}{\sqrt{l^2-r^2\sin^2\theta}} - \frac{r^4(\sin^2\theta\cos^2\theta)}{(\sqrt{l^2-r^2\sin^2\theta})^3}$

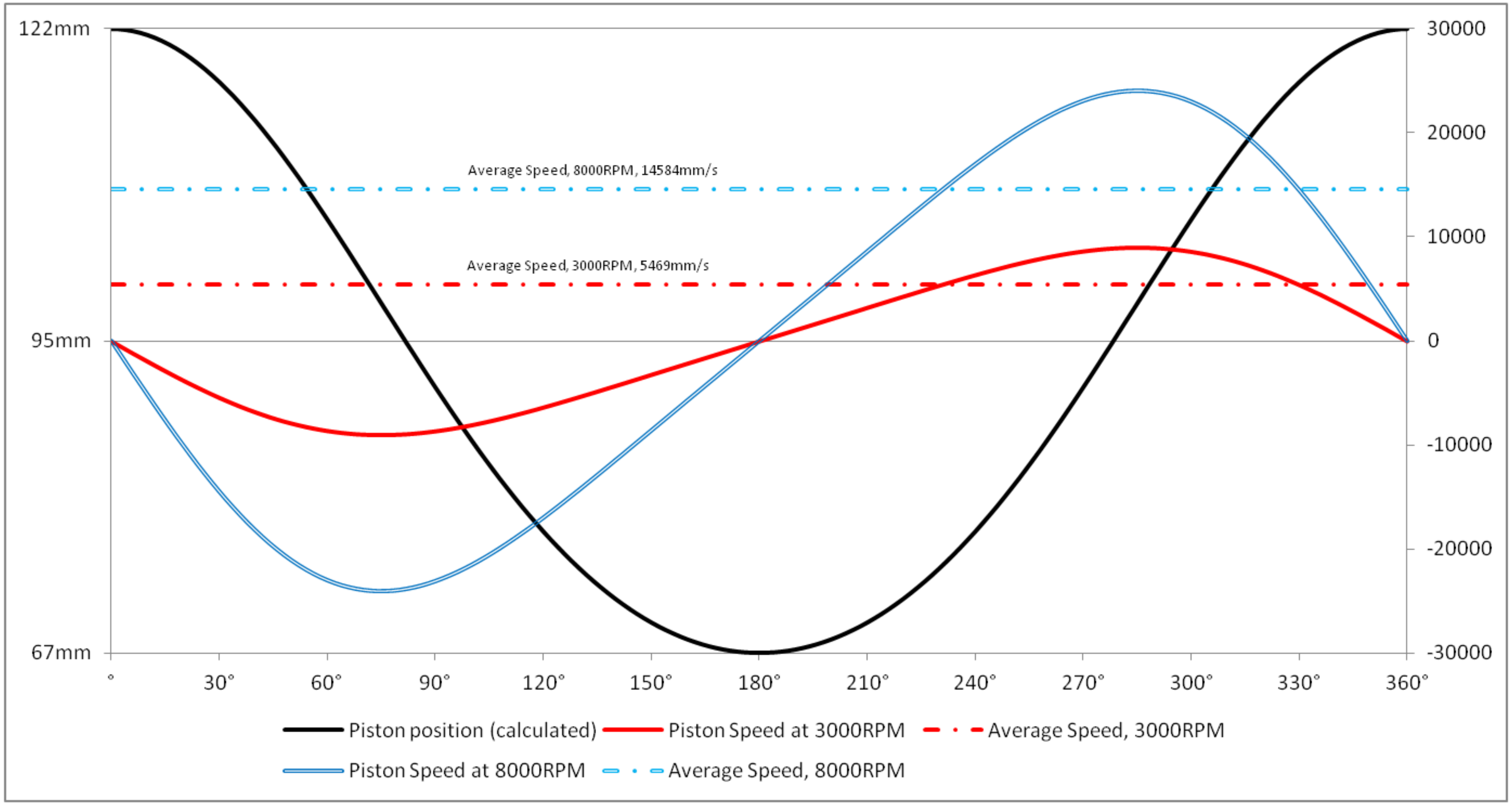
From the above equations, and computing over a range of 360° of Crank Angles, the following graph is obtained. The calculations suggest a deviation from the original sinusoidal assumption of the motion of the Piston, as shown in Figure 10.

Based on the previous calculations in which the graph obtained was with respect to the angular displacement, additional calculations can be done to illustrate the piston speed and position with respect to time, calculated for various RPMs. The result is shown in Figure 11.



Vertical Axis 1: Piston Position (mm)   Vertical Axis 2: Piston Velocity (m/s) and Acceleration (m/s²)
Horizontal Axis: Crank Angle (degrees)

Figure 10: Piston Position, Velocity and Acceleration over 360-degree Crank Angle



Vertical Axis 1: Piston Position (mm) *Vertical Axis 2: Piston Velocity (mm/s) and Acceleration (mm/s²)*
Horizontal Axis: Crank Angle (degrees)

Figure 11: Piston Speeds at 3000RPM and 8000RPM

From Figure 11, it can be seen that the piston speeds are affected by the RPM of the engine, where a higher RPM will require a faster piston speed to match. The calculations also coincide with previously calculated *Mean Speed of Piston* for 3000 RPM, where the average speed plotted out here is obtained by calculating the RMS value of the piston speeds over the entire 4-stroke cycle.

Combining the information about the piston and intake valves, a model of the vacuum created at various RPMs can be formulated, possibly to input into a flow simulation, to represent the effect of the falling piston creating a vacuum in the cylinder and propagating that vacuum pressure through the rest of the air intake system.

INTAKE RUNNERS

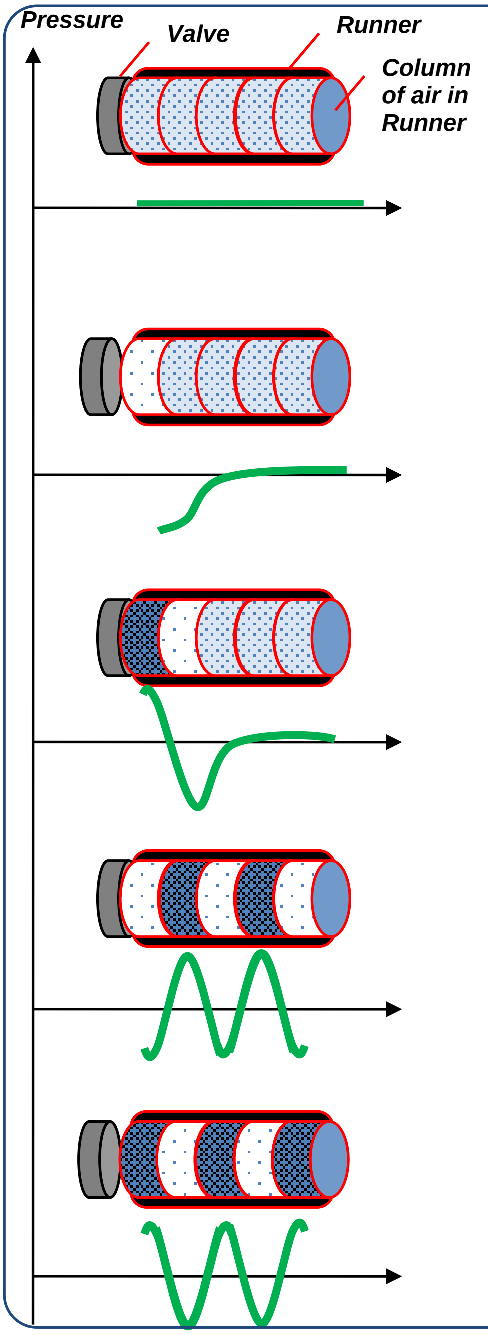
The intake runners are the parts of the air intake system which delivers air from the intake manifold to the cylinders (see Figure 3). In each runner, the dominant phenomenon that governs its performance is actually the effect of acoustic waves. As the purpose of the runner is air delivery, its performance is pegged to how much air it can deliver, and in the case of the engine, the subsequent improvement in volumetric efficiency.

Air Ramming using Tuned Intake Runners

The behavior of acoustic waves in the runners resembles sound waves travelling through an open tube. An acoustic wave is a longitudinal wave made up of alternating regions of compressions and decompressions. In an intake runner, similar compressions and decompressions are created.

When the intake valves first open, the suction created within the cylinder will generate a region of low pressure at the cylinder end of the intake runner. Around the middle of the intake stroke, when the piston is moving at the maximum speed downwards, would be when the suction or decompression reaches a maximum. Towards the end of the intake stroke, the decompression will start to reduce, until it totally stops when the intake valves close.

However, the inertia of the air column inside the intake runner causes it to continue moving towards the intake valves, and thus, having nowhere to go, the air column starts stacking upon itself, creating a region of higher pressure against the back of the intake valves, otherwise known as a region of compression. With this alternating compression and decompression waves being created, an acoustic wave is formed. These regions of high and low pressures are being propagated away from the intake valves as they are being created.



1. ***Air in the runner is initially stagnant, at atmospheric pressure, and evenly pressured throughout.***
2. ***As the Intake Valve is opened, vacuum created by the downward (towards left in this diagram) moving piston creates a negative pressure region at the back of the Valve.***
3. ***Once the Intake Valve is closed, the air column that was flowing into the cylinder rushes against the back of the Valve with residual inertia, creating a region of high pressure in the runner, behind the Valve.***
4. ***This alternating high-lo pressure is being propagated towards the manifold end of the runner (the right side of the cylinder) where it is reflected back.***
5. ***The objective of a tuned intake runner would be to manipulate this reflected wave (while the engine is running at a desired RPM) to make sure that a high-pressure region coincides with a subsequent Valve opening, improving air-flow into the cylinder, improving volumetric efficiency, hence achieving peak torque at the required RPM.***

Figure 12: Pressure waves created by intake valve opening and closing

On the opposite end of the runner, which is connected to the intake manifold, the sudden change in cross-sectional area of the runner will appear to be a “wall” for the propagating wave, terminating the acoustic wave’s propagation, and thereby creating a reflection of the wave towards the intake valve.

This phenomenon is always present, regardless of which RPM the engine is operating at. The objective of a good runner design would be to “tune” it such that at the RPM targeted for peak torque, there is a reflected, high-pressure column of air waiting behind the intake valves, ready to be sucked into the cylinder.

By increasing the pressure of the air column waiting behind the intake valves, it is similar, to a small extent, to super-charging the engine, forcing a larger quantity of air, albeit at the same volume, into the cylinder. This effectively raises the volumetric efficiency of the engine at that point, consequently increasing the power output as well. At the RPM point which the intake runners are tuned for, there will be a maximum volumetric efficiency, and equivalently a maximum power output, with this point aptly termed as the peak power RPM point.

In order to achieve this, the higher-pressure column of air, which starts forming upon the closing of the intake valve, will have to propagate to the open end of the runner, be reflected, and propagate back to the valve opening within the duration of the intake valve’s closure. Knowing the time required for the distance travelled, as well as taking the assumption that this acoustic compression and rarefaction wave propagates at the speed of sound, a simple calculation can be done to obtain the runner length to accommodate such a distance. Having an intake runner sized at the appropriate length to increase the pressure of the air behind the intake valves when they open, is known as runner length tuning. A properly tuned intake runner system will be able to “ram” more air into the cylinder and thus improve the overall volumetric efficiency.

Selecting a Length for the Intake Runners

The calculation for the length of the tuned intake runners is based upon the duration between two events of the intake valve’s operation: the first being the closing of the intake valve, and the second being the re-opening of the intake valve on the next cycle. From the Engine Datasheet in Appendix A, the information is found to be 48°ABDC (After Bottom Dead Centre, referring to the angle the crankshaft rotates after passing the BDC position) to 14°BTDC (Before Top Dead Centre, referring to the angle the crankshaft needs to rotate before reaching the TDC position) respectively, giving a total of 474° of crankshaft rotation during which the valve is closed. Further calculations factor in the effect of the engine speed, such that various RPMs will create different time durations within which the intake valves remain closed.

Taking a full 4-stroke cycle to be made up of two crank cycles, the intake valve is thus closed for 474° out of a total of 720°, giving a duty-ratio of 65.8%. Therefore, considering the RPM and how an increasing RPM will suggest a reducing time period for the wave to reflect, a range of values is calculated for the lengths of runners required to achieve the effect of reflecting the higher-pressure air onto the intake valves are they open.

Number of 4-stroke cycles per second = $\frac{RPM/60}{2}$
Crank Position @ Intake Valve Closing = 48° ABDC
Crank Position @ Intake Valve Opening = 18° BTDC on next cycle
Total Crank Angles during which Intake Valve is closed
= (180° - 48°) + 180° + (180° - 18°) = 474°
Duty Ratio of Intake Valve Closed time = 474° / 720° = 65.8%

Figure 13 below shows the result of these calculations, expressed as lengths (in cm) against RPM values. The initial observation is that in order to obtain the ramming effect discussed above, at a low RPM of about 2500, an intake runner length of over 5m would be required! This would be close to impossible to package in the FSAE car, as well as to justify the extreme increase in weight for the relatively little improvement in volumetric efficiency.

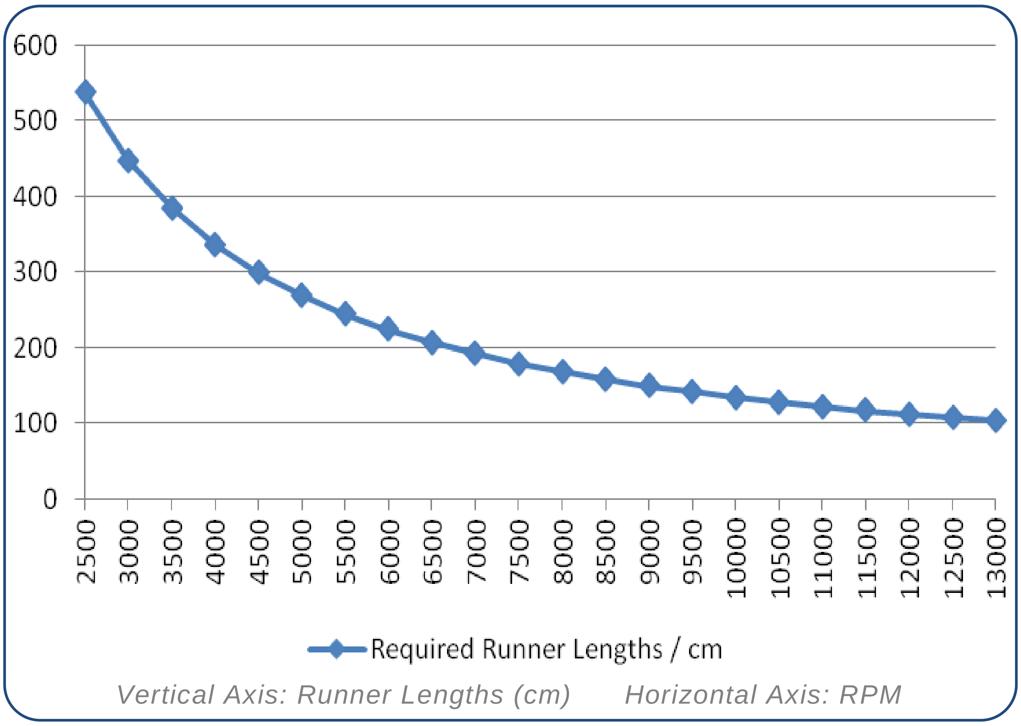


Figure 13 : Projection of Required Runner Lengths

The compromising solution to the problem would be to make use of the higher order harmonics of this phenomenon, which suggests that while there is a preferred length to create this harmonic effect in the runner, a shorter runner can actually be employed to create a similar harmonic effect, except that instead of travelling the length of the runner only once, back and forth, the created higher-pressure air column will reflect back and forth for a few cycle, between the back of the closed intake valves as well as the open end of the runner, and would eventually position the higher-pressure air column back to the opening intake valves, improving the air intake, and consequently the volumetric efficiency of the system.

The trade-off for using a shorter length runner is that as the acoustic waves propagate back and forth, they will constantly lose energy, so as the runner is designed to accommodate a higher order harmonic, and therefore designed to be shorter and easier to package on the FSAE car, it is also suffering the effects of a decreased ability to improve volumetric efficiency through the ramming effect of the acoustic waves.

Figure 14 below shows, in a three-dimensional expansion of the previous plot in Figure 13 , the lengths required for intake runners. The plot is done over the same range of RPMs, only that it includes an additional axis this time to encompass the various harmonics of the runner lengths. The plot above shows a calculation of the required runner lengths if the various harmonics were to be adopted. For example, attempting to achieve this ramming effect at 2500RPM, using the 8th harmonic (the acoustic wave travels up and down the runner's length 8 times between the intake valve closing and opening), would require an intake runner length of about 80cm, which is still considerably long, taking into account the small size of the race car.

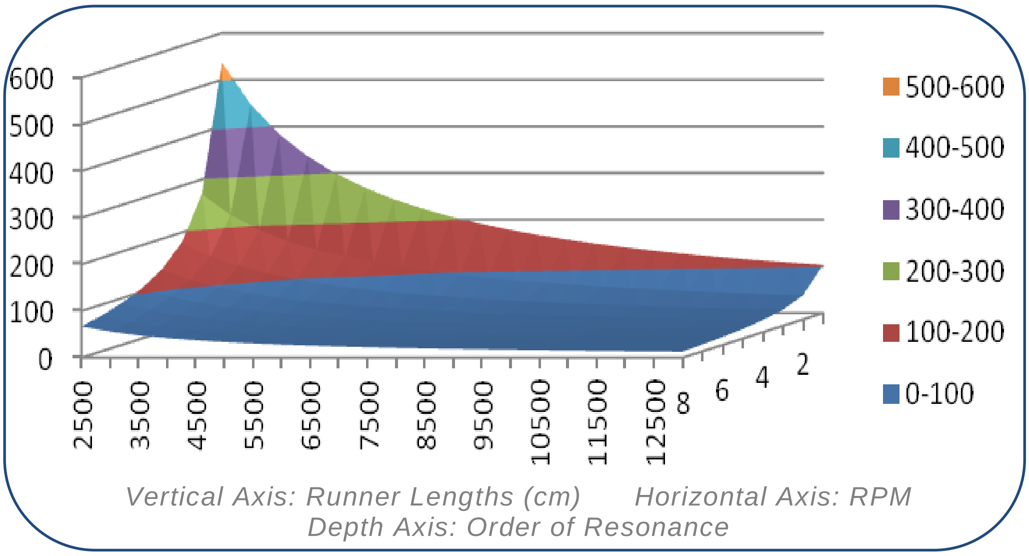


Figure 14: Required Runner Lengths up to 8th Harmonic

In Figure 14 above, the requirement of an intake designed for low RPMs, making use of the first few harmonics, would still exceed 200cm, equivalent to over 2m of piping, an impractical length of tubing to be located within the FSAE car. Due to space constraints, it is estimated that the probable suitable lengths of the intake runners should lie between 10 ~ 40 cm, which was how the following graphs in Figure 15 had been derived.

Figure 15 shows the lengths required to take advantage of the various resonance frequencies at a range of RPMs, up to a maximum length of 40cm. From the upper graph in Figure 15, it can be seen that many of the first-order to third-order harmonics are not usable due to the extremely long lengths required of the runners. A compressed view of the graph (lower graph of Figure 15) shows the required runner lengths, in colored bands, for the various RPMs and their respective orders of harmonics.

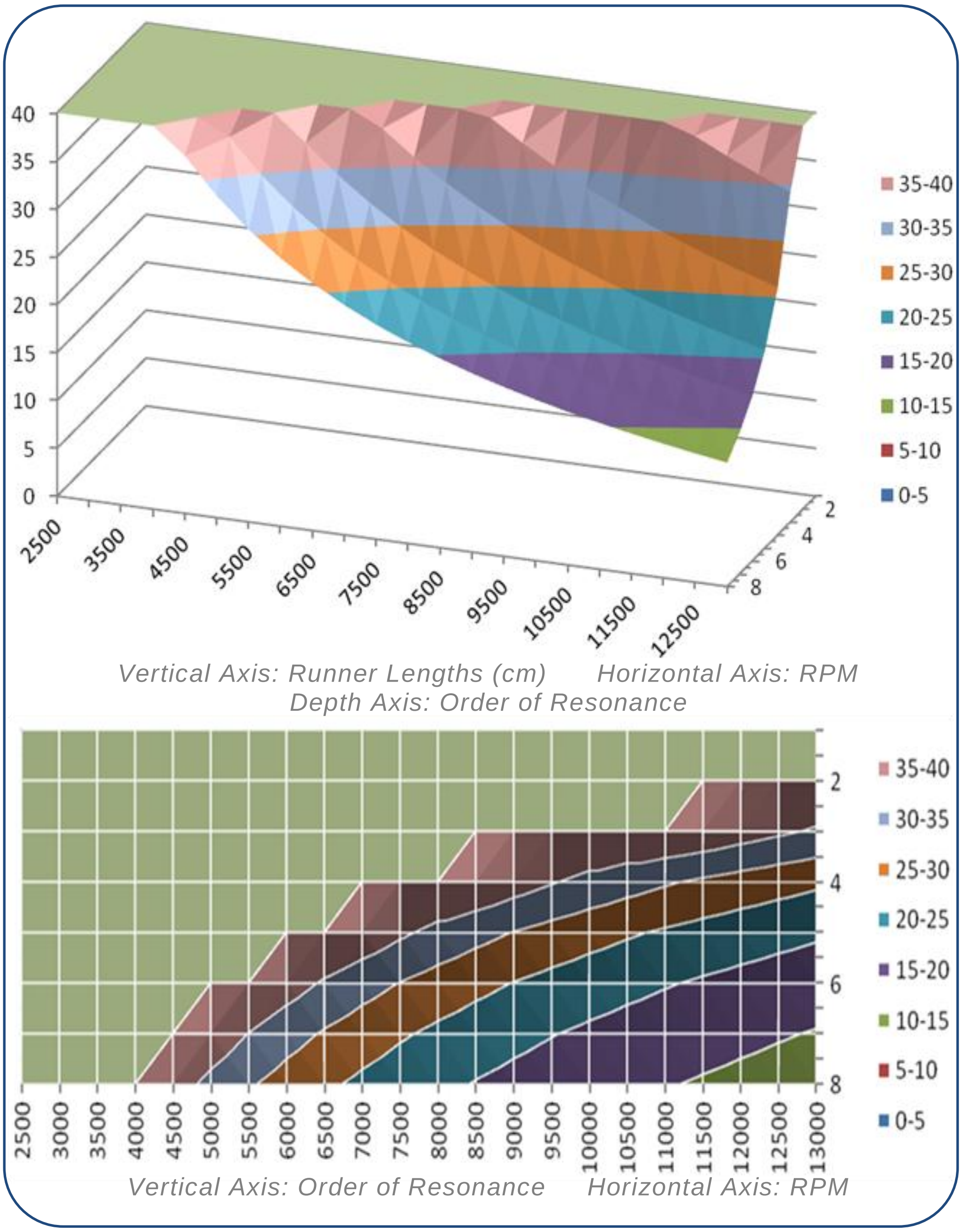


Figure 15: Runner Lengths, up to 40cm

The calculated results suggest that, for example, if the targeted power band was to be between 4000 to 5000 RPM, runner lengths between 35 – 40cm will only be able to provide a ramming effect at the 8th order, or if the allowable length of the runner was to be between 25 – 30cm, using the 5th order harmonics would provide an air ramming effect for

9000 to 10500 RPMs. These values would give an approximate starting point to the design of the intake runners.

Detrimental Effects of Tuning Runner Lengths

While the tuning of runner lengths give the engine the capacity to improve its volumetric efficiency by timing a higher-pressure air column into the cylinder when the intake valves open, it would be necessary to note that the length of the runner is only tuned for a particular RPM, and that at other RPMs, particularly those at which the reflected waves position a lower-pressured air column at the intake valves as it open, will suffer a decrease in volumetric efficiencies as a result.

Therefore, while runner lengths increase the power at the designed RPM, it will create reduction in power at others. In the case of designing for a peak power at a fixed RPM, all the design components in the engine system can be specified to target that RPM. However, it is unlikely that the engine stays at a fixed RPM, and therefore a flatter torque curve would be beneficial to the tolerance of the race car for an amateur driver, in terms of throttle and engine control. This is done by varying the design objectives of various components in the design, such as the runner lengths and the primary and secondary exhaust lengths, to create targeted peaks at different points of the RPM range, resulting in multiple peaks that superpose into a flatter but lower torque curve.

INTAKE MANIFOLD

The intake manifold, also known as the plenum, or the air-box of an engine, is a reservoir of air from which the engine cylinders will draw from. This air is, in turn, replenished from the atmosphere through the restrictor, throttle body and intake trumpet.

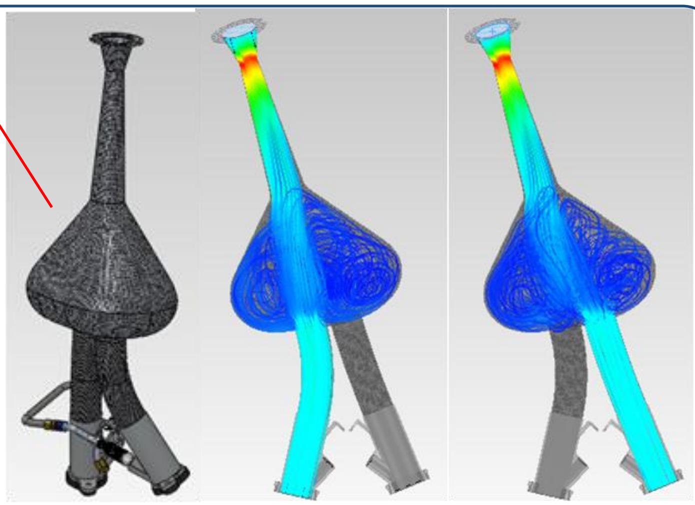
In an intake system supplying two or more cylinders, and particularly one with a restricted intake system, as is the one in the FSAE car, it is essential that there is a manifold that is able to supply the runners with a sufficient flow of air. Individual runners will be able to draw from the reservoir of air in the manifold, so that there will not be a situation where the cylinders mutually starve each other.

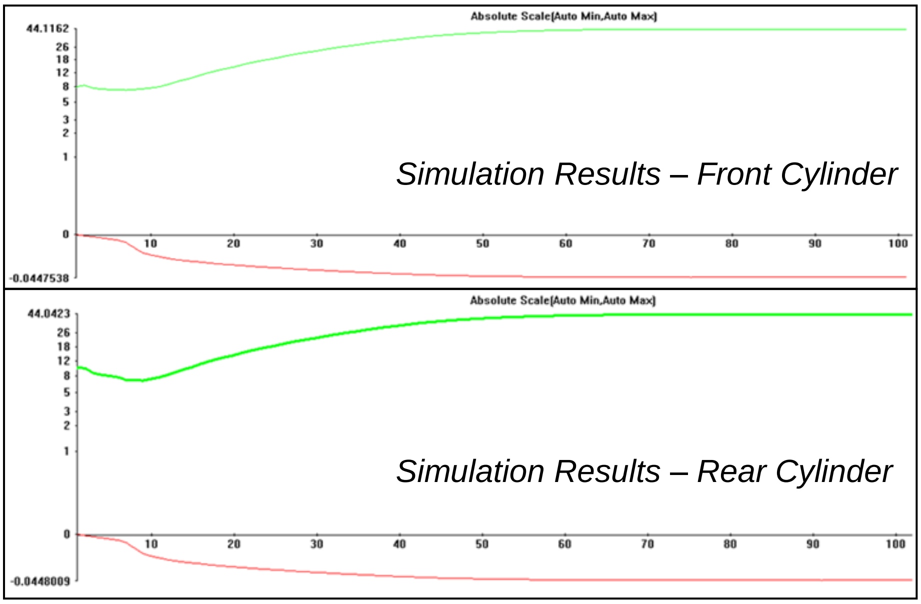
The manifold also seeks to reduce the impact of imbalanced distribution to the individual cylinders, in some cases being geometrically asymmetrical, creating a pressure gradient over each cylinder's runner inlet, such that the volume of air entering each cylinder can be consistent.

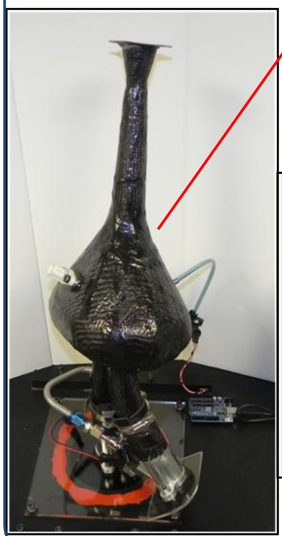
Even Distribution of Airflow

As the team has generally been operating on multi-cylinder engines, one of the chief roles taken up by the manifold is to evenly distribute the air into various cylinders. This is critical in multi-cylinder engines to ensure that there is even combustion across the cylinders so that the engine is more stable and predictable.

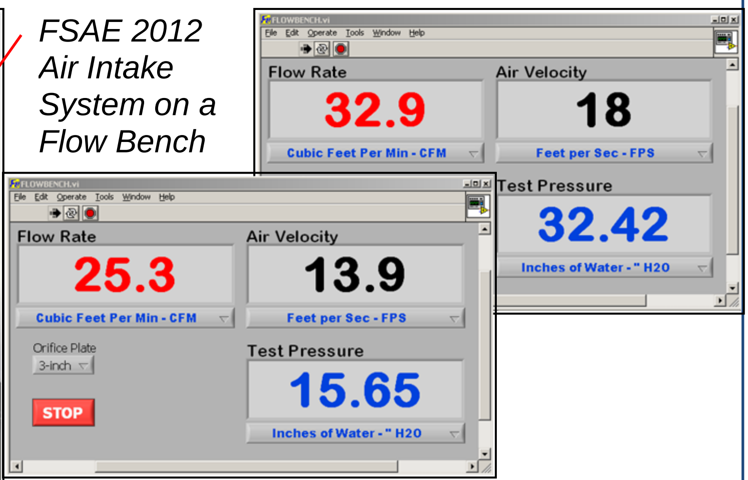
Simulation Model of FSAE 2012 Air Intake System with Flow results for front and rear runner's air distribution (with pressure distribution)







FSAE 2012 Air Intake System on a Flow Bench



Sample Flowbench Results

Figure 16: Testing the Intake Plenum for Evenness of Air Distribution

Cylinder	FRONT	REAR
Simulation Flow Rate, m^3/s	**44.12**	**44.04**
Simulation Air Velocity, m/s	**0.045**	**0.045**
Flowbench Speed I Flow Rate, CFM	13.37	13.17
Flowbench Speed I Air Velocity, FPS	7.31	7.21
Flowbench Speed I Test Pressure, "H_2O	15.63	14.85
Flowbench Speed I MAP Sensor, V	**2.82**	**2.83**
Flowbench Speed II Flow Rate, CFM	20.63	21.37
Flowbench Speed II Air Velocity, FPS	11.31	11.68
Flowbench Speed II Test Pressure, "H_2O	31.95	32.18
Flowbench Speed II MAP Sensor, V	**2.75**	**2.72**
16"H_2O Normalized Speed I Flow Rate, CFM	**13.52**	**13.67**
16"H_2O Normalized Speed I Air Velocity, FPS	**7.40**	**7.48**
16"H_2O Normalized Speed II Flow Rate, CFM	**14.60**	**15.07**
16"H_2O Normalized Speed II Air Velocity, FPS	**8.01**	**8.23**

In order to verify the evenness of the air distribution, several tools can be used. Firstly, a simulation can be carried out to check each iteration of the design. From Figure 16, as well as the table above, it can be seen that the simulations have produced desirable results of similar flow rates and air velocities for both the front and rear cylinders, as well as even pressure distribution when the flow occurs. Flow simulations are discussed further in Chapter 4.

When the plenum is eventually manufactured, it can be connected to a Flowbench, and the actual evenness of the air distribution can be verified. In the table above, it can be seen that most of the figures are closely similar between the front and the rear cylinders. The tests were carried out using the two available speeds on the equipment, and subsequently normalized to 16"H_2O test pressure.

Also, during the flowbench tests, the Manifold Air Pressure (MAP) sensor is being monitored, and portrayed similar output voltages. This suggests that the pressure in the manifold during the tests on the front and on the rear runners are almost the same, leading to the conclusion that an intake manifold designed like this is able to evenly distribute airflow to either of the two cylinders it provides for.

Shapes of the Intake Manifold

Another purpose of the Intake Manifold is to act as a buffer for the air flow. The most apparent example of this is in a super-charged engine, where an air pump constantly creates positive pressure as it is driven. However, the cylinders are constantly in a state of motion, and the resultant demand in air supply constantly changes. Without a buffered reservoir of air to draw from, the cylinders will not be able to breathe in a predictable manner, making the drivability and usability of the engine poor. Coming back to a normally aspirated engine, in order to maintain a predictable engine behavior, an intake manifold needs to be able to even out the incoming flow of air, such that the suction at the corresponding runners of each cylinder is even. Besides restrictions in the physical envelope of space, this would be one of the two main determining factors of the shape of an intake manifold.

The other determining factor of the intake manifold shape is the pickup location of the intake trumpet in reference to the car. Typically, the location would avoid any local low pressure points created by the aero dynamic package of the race car, as well as avoid areas where heat can be trapped, such as around the exhaust headers or behind the radiator. For the configuration of the FSAE race cars, this usually leaves three primary areas to start the air intake system, along the longitudinal axis of the car near the top of the main roll hoop, towards either sides of the car slightly above the cockpit's height, and finally a rearward facing intake point near the rear box of the race car. The various examples of each are shown in the following Figure 17:

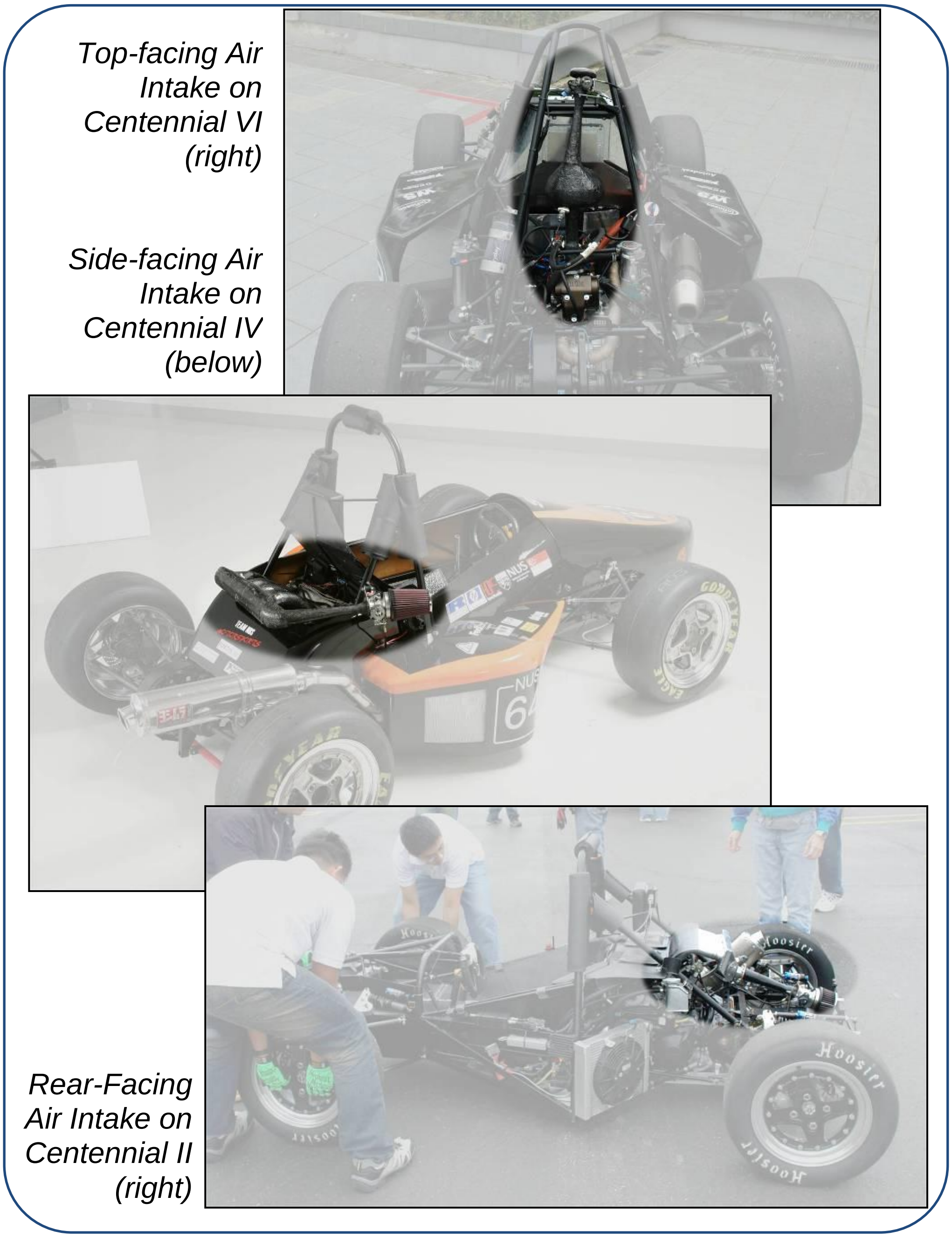


Figure 17: Various Air Inlet Locations

For a basic, non-variable-length, normally aspirated configuration, this usually suggests one of two general shapes, the “onion” or the “log”. In a “log” shape, particularly useful for inline-four-cylinder engines, one of the major considerations would be flow distribution, which usually means that there is a narrowing of the intake manifold from the point where the air inlet enters through the restrictor, to the furthest point from that air inlet. This accounts for the trapezoidal shape from a centre-fed log manifold in the Centennial II and a triangular shaped side-fed log manifold on the Centennial IV. Using a V-twin engine, where there are only two intake ports that are much closer together, an “onion” or a spherical-like shaped is used.

Based on limitations of the other components in the air intake system determining the entry and exit points of the manifold, adjustments to the shape can be made through iterative flow simulations and analysis, targeting for a manifold that can allow the air to settle before entering the runners, as well as to ensure that the cylinders are evenly-fed air.

Size of Intake Manifold

After determining the approximate shape and configuration of the Intake Manifold, another key feature to decide upon is its size. This size refers to the internal volume of the intake manifold.

The size of the intake manifold directly affects the performance of the engine, but is a component that is difficult to determine the suitable size for. Besides the point of packaging the intake manifold, arguable the largest component of the air intake system, within the envelope of the FSAE race car, it has to satisfy both the steady state and transient requirements of the engine.

On the one hand, the intake manifold volume must not be too small. It is required that the intake manifold be at least the size of the engine capacity, such that it holds enough air to provide for the cylinders during each cycle. It is also recommended that the size be at least two times to allow the engine to be able to draw air while maintaining a stable pressure in the manifold. Tests have, however, determined that for a steady state analysis, the larger the intake manifold, the better, as it increasingly approximates to an open atmospheric environment from which the engine can draw air from.

However, while the static analysis of the airflow suggests that the manifold be as large as possible, an increasing large manifold has a detrimental effect on the throttle response of the race car, considering that the throttle body has to be located prior to the intake manifold. An example of this effect is that the pressure in a manifold that is very large will change very slowly, with respect to the change in the throttle position. On a race car, where the driver changes his throttle inputs at very quick rates, it would not be acceptable.

The size of the intake manifold, must therefore, be a compromise between a size large enough to provide the individual cylinders with sufficient air, but yet not too large that throttle response is compromised.

INTAKE RESTRICTOR

The restrictor is a component mandated by the rules of the competition, in which all the air entering the engine must pass through this 20mm diameter gap as a way to restrict the engine's power. With a restrictor placed early in the air intake system, engine performance is being greatly compromised, as it is proportional to the volumetric efficiency of the engine system, which is in turn related to the amount of air which can be drawn in by the cylinders.

It is therefore critical to design the restrictor to ensure that the maximum airflow can be passed through the restrictor, so as to allow the cylinders to take in as much air as possible, during the intake stroke. This will allow the maximization of volumetric efficiency across various RPMs.

Shape of Restrictor

The shape of a restrictor can be as simple as a plate with the mandated dimensions machined into it, and placed anywhere along the air intake system. However, a simple orifice plate would create a lot of pressure loss downstream of the restrictor, and the resultant effect is an increased inefficiency along the line of airflow.

In observing the flow of air through an orifice plate using a CFD tool, it can be seen that there is a point in which air converges beyond the orifice plate, known as a *vena contracta.* This convergence leads to a restriction of air flow that is smaller than the orifice plate opening itself. Therefore, there is a need to guide the air into the restrictor, instead of creating a sudden step that would have been in the case of the orifice plate. Thus came the common design of a Convergent-Divergent Nozzle that replaces the orifice plate as an air intake restrictor.

The CD Nozzle is a tube which, on one end, is exposed to the environmental atmosphere, tapers into the mandated restriction diameter, and then tapers out into the manifold chamber, and in doing so seeks to reduce the pressure loss across its length as much as possible.

In many documented studies and literature, the recommended shape for the convergent part of the restrictor is an elliptical curve leading to the minimum diameter point, while a 3° to 7° taper on the divergent end of the restrictor would allow the air to regain the pressure lost as air flows into the constriction.

Figure 18 below shows the effect of flow through an elliptical convergence and 4° divergence CD-Nozzle as a restrictor:

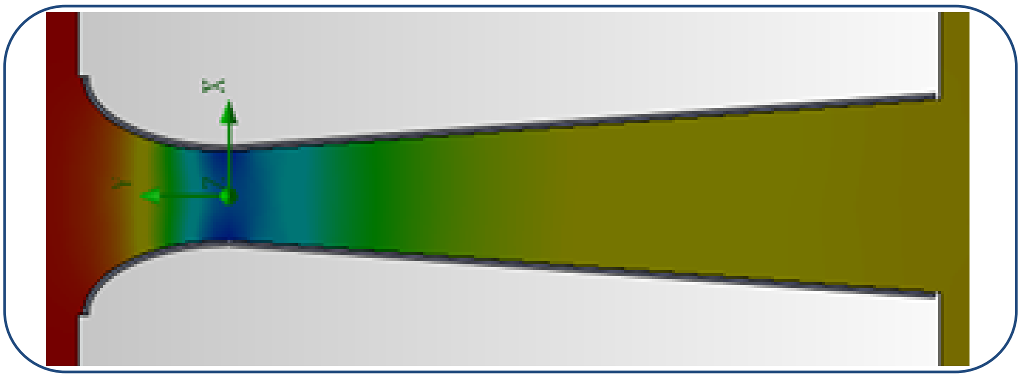


Figure 18: Flow through a CD Nozzle

THROTTLE BODY

The intake pick-up point on a Formula-SAE race car is the first and foremost engine component that is in contact with the atmosphere. Air will enter the intake system and the engine through the inlet. Three components make up the intake pick-up point sub-system, specifically the air filter, intake trumpet, as well as the throttle mechanism.

Types of Throttle

The throttle is a valve placed at the entrance of the air intake system, and is the primary means to control the behavior of an engine. Connected directly to the throttle pedal via a steel braided cable (due to restrictions against drive-by-wire systems), and placed ahead of the air restrictor (as mandated by the competition rules), the throttle body is the first and foremost component between the air entering the engine and the atmospheric air.

For such a critical component, there are several variants which can be used to achieve a variable air inlet, and the three primary designs that have been adopted are the Barrel throttle, the Slide throttle and the Butterfly throttle, all with their individual pros and cons.

The Barrel throttle is similar to the mechanism inside a water faucet, and one of its main advantages is that at wide-open throttle, it has a “clear barrel”, indicating that the flow through the throttle will be totally unobstructed. The trade-off for such a throttle, however, is the complexity, size and weight of the mechanism required to support it.

The next throttle that is commonly seen is the sliding throttle, in which a plate matching the opening of the throttle body is being moved across the hole, modifying the path of air flow from a small slit, to a clear, un-obstructed clear barrel at maximum throttle. In comparison, the mechanisms required to build this throttle body is relatively lesser, and smaller, but it has a major design flaw, which is a difficulty to return the sliding plate when air is rushing into the intake manifold. When the engine is at a high RPM, after leaving the throttle wide open for a moment, the speed of the incoming air will push against the sliding plate to prevent the throttle from closing, posing a grave threat to the driver who might have required the car to slow down.

In a solution to the problems of both of these throttle systems, a butterfly valve throttle has become one of the most commonly used mechanisms in the modern day engine. The mechanism is much smaller, requiring only a small housing clamped around the throttle itself to hold the mechanisms. This configuration also proves to be a lighter solution. In addition, as the mechanism is already in line with the air passage, there are little problems in closing the mechanism even while at wide-open throttle. However, this mechanism is not without its problems, as the mechanism is located inside the throttle body itself, leading to a problem of having an obstruction in the path of the air flow, even at wide open throttle.

However, the work-around solutions for this particular mechanism is considerably simpler when compared to the barrel and the slide throttles. One of the solutions is to increase the dimensions of the throttle body diameter slightly, in order to compensate for the reduced cross-sectional area during the wide-open throttle. However, the main solution for this is so make the throttle plate inside the butterfly valve as aero-dynamic as possible, reducing the profile that obstructs airflow.

The following images in Figure 19 show the effect of a butterfly throttle body in (A) as compared to a clear barrel in (D). it also shows the effect of optimizing the throttle plate aerodynamically to reduce the effect of impeding air flow, and with more optimization, the presence of the butterfly valve can still approach that of an open barrel, shown by sharpening the plate in (B) and trimming down the shaft holding the plate (C) .

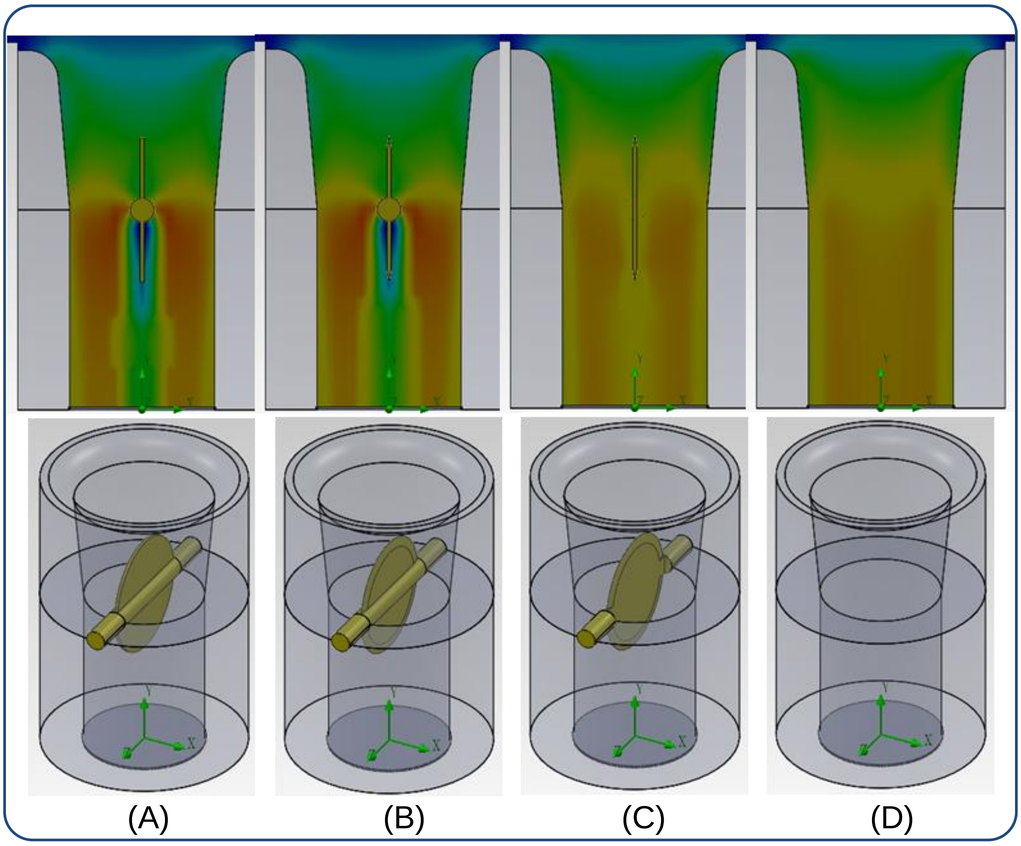


Figure 19: Effect of Throttle Plate and Design Optimization

Effect of Air Filter

Another critical design of the Throttle Body would be the type and size of Air Filter to use when running the FSAE race car. Under operating conditions, the FSAE race car kicks up many debris particles from the ground over which it runs, ranging from dead leaves to sand and even small rocks. Those, in addition to other foreign airborne particles, are harmful additions to the air in which the engine is taking in. In the best case, these particles are disintegrated when they enter the combustion chambers, but decrease the power created. In the worse case scenario, these particles accumulate and eventually cause damage to the pistons and combustion chambers, and lead to piston seizure, and eventually to a blown engine.

There is, therefore, a need to run an Air Filter ahead of the Throttle Body to prevent these particles from entering the Air Intake System, and eventually the engine. However, it becomes apparent that a filtration system, having only small openings for air to pass through, would therefore become and obstruction to airflow, a restrictor in itself. It is therefore necessary to test the flow behavior of various air filters in order to choose a suitable one, balancing its size, weight and performance.

For this segment, various brands of air filters can be purchased and tested for a tabulation of their flow performance and physical characteristics, but will not be covered in this study. However, establishing the procedure through which the air filters can be tested was carried out, and a flow test was carried out, using the Flowbench, to determine the effect that the current Air Filter that the FSAE team is using, has on the intake system's flow rates. The test, shown below, in Figure 20 was carried out using the 2012 Throttle Body, with and without the Filter. Flowbench testing is further explained in Chapter 4.

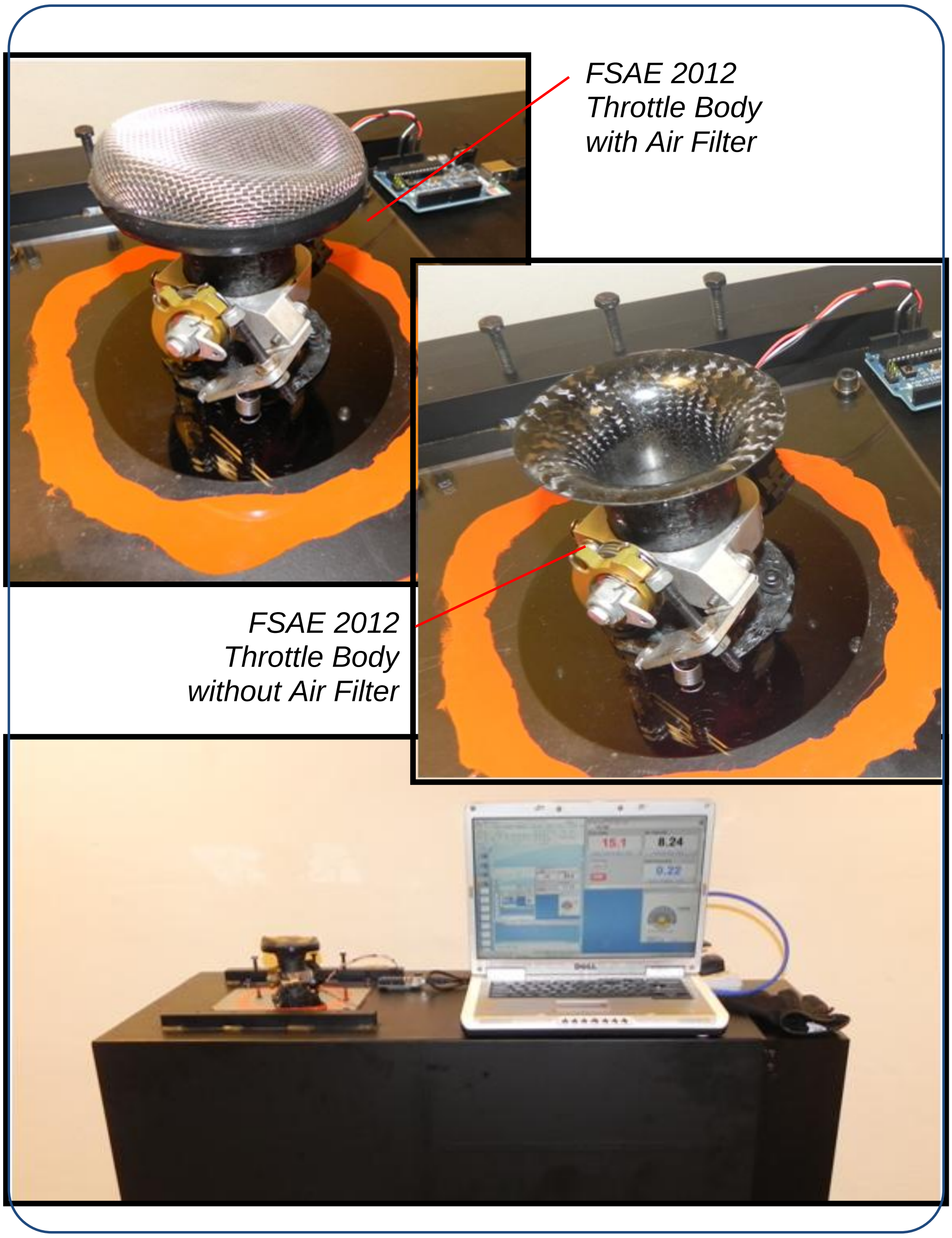


Figure 20: Flowbench Testing for Effect of Air Filter

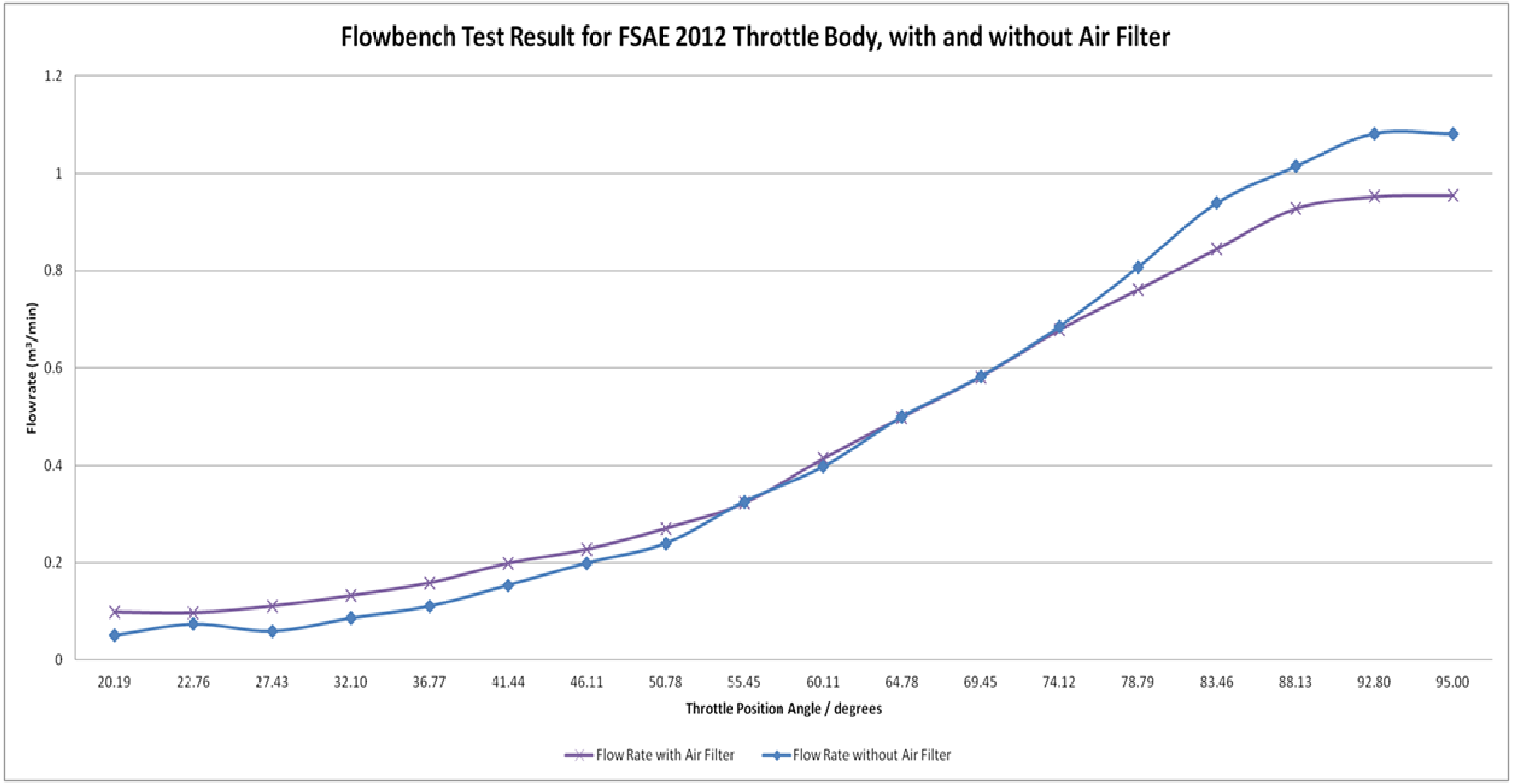


Figure 21: Effect of Air Filter on Flow Rate

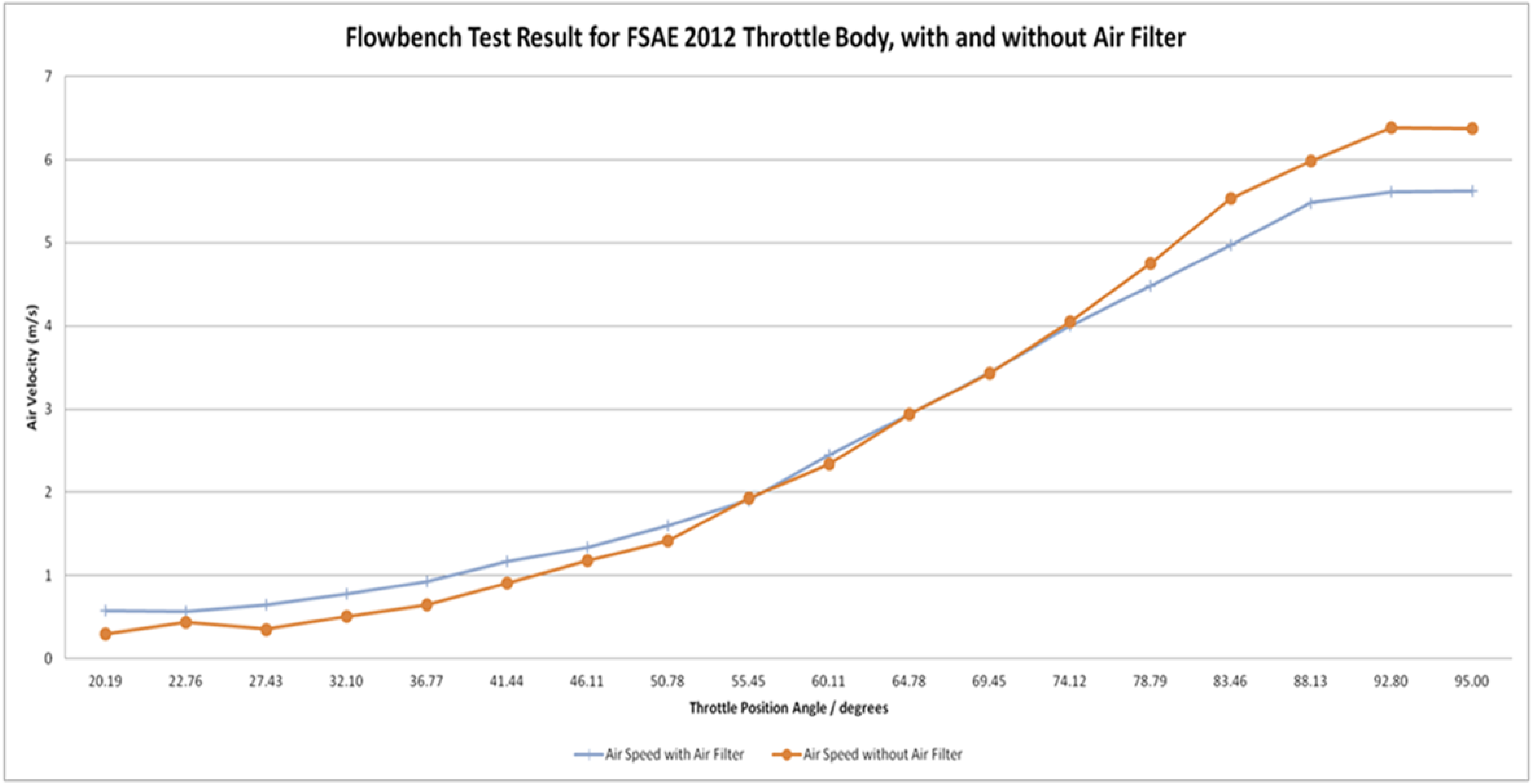


Figure 22: Effect of Air Filter on Air Velocity

Figure 21 and Figure 22 show the results of flowbench tests on the 2012 throttle body with and without the air filter. It can be seen that at lower throttle positions, where there is a lower flow rate and air velocity, it is actually observed that values are actually higher with the air filter. This departs from the original expectation of a reduced volumetric air flow and velocity across the entire range of throttle positions.

A possible hypothesis for this phenomenon is that at lower speeds, the filter elements of the air filter, which is made up of a series of wire meshes, could actually be aligning the flow of the air entering the cylinder, creating an area of laminar flow. At the same low-throttle-opening position, the throttle body without an air filter is actually drawing air from all directions, creating a turbulent flow area at the centre of the throttle body opening, reducing the air flow through the throttle body.

On the other hand, as the throttle opening increases with the increasing angle of the throttle plate, the volume flow and air velocity both start to increase, to the point where it is able to generate laminar flow through the throttle body. Beyond this point, where laminar flow is dominant across the throttle body and the effect of turbulent flow impeding the air flow is much less significant, the effect of the filter elements being an obstruction to airflow becomes increasingly significant.

This observation had drawn attention to the importance of accounting for airflow that is not simply flowing straight into the intake component, but also the flow from all angles, possibly entering the intake components from all angles and the creation of turbulent flow at the opening of the intake. It had thus sparked off an iteration of modeling the airflow through these components, in which the model for the environmental air is being expanded from simply a cross-sectional area at the opening of the system, to a significantly larger environment to simulate the possibility of air entering the components from side-wards angles. This improved model is documented in the section named "Improved Model for Environmental Pressure Simulation" on page 60.

Simulation was also carried out to explore the effect of the air filter using a CFD perspective. While it was not able to simulate exactly the air filter's characteristics, the placement of a "porous medium" over the simulated throttle body aims to be a usable model of the air filter material. The simulations models used are shown below in Figure 23:

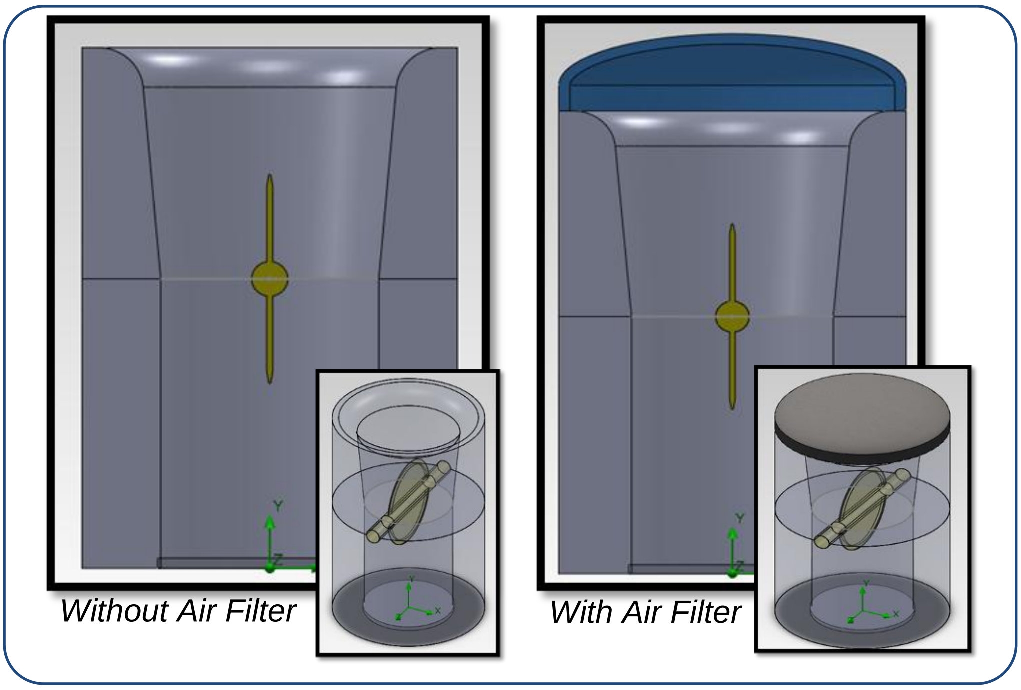


Figure 23: Model of Throttle Body with and without Air Filter

The simulations were then carried out with flow being done in one of the three conditions: without an air filter, with a screen-type multi-directional porous material as a filter, and finally with a uni-directional (in the axial direction of flow through the throttle body) porous material as a filter. The results of these tests are documented in Appendix B on Page 81.

From the results shown, one of the clearest features is that the presence of the filter is not as significant at lower throttle positions, but increasingly significant changes can be observed as the throttle angle increases. Another observation is the limitation of air velocity by the air filters, mirroring the effect observed on the flowbench tests.

4. SIMULATION AND TESTING

SolidWorks Flow Simulation

SolidWorks Flow is a powerful yet simplified add-on to the SolidWorks CAD software that allows a user to quickly carry out a Computational Fluid Dynamics (CFD) analysis on a certain assembly. It comes with a user-friendly Wizard that allows the user to set up the fluid flow and fluid forces analysis.

In this segment, the SolidWorks Flow software is used to analyze the airflow in the air intake system. It allows the user to check on the feasibility of the designed component and carry out an initial evaluation on the feasibility and quality of a certain design. There are various forms of flow analysis, mainly being divided into Internal and External Flow. For the purpose of the air intake system, the Internal Flow analysis is being employed. It is chiefly the study of a fluid's flow within a volume contained within a solid boundary, in this case the geometry of the air intake system. As with most finite element analysis, computing power is inversely proportional to the time required for computing, while simplifying the model will provide faster times for meshing and running.

Preparation for Analysis

Before an analysis can be carried out on the modeled air intake system, certain preparations need to be made to the CAD model. The primary requirement of the model is that it must be completely closed, and the volume through which the air is supposed to be simulated to flow through, should be a completely enclosed surface. This means that, beginning from the point of air inlet, which in the case of an entire air intake system will be the location of the intake pick up point, all the way to the air outlet, which is where the intake runners connect to the engine, there should be no gaps or holes in which air can enter or exit the system to be simulated.

The most straightforward way of doing this is to create “lids”, or imaginary solid covers that cap the entrances and exits of the system to be tested. Subsequently, the lids will be used to create boundary conditions, and determine the starting conditions of the interfaces between the test volume and the environmental non-tested volume.

In addition to preparing the model, Goals can be added, whether it is at a specific surface, or on a global scale (goal for entire model). Setting goals will force the simulation engine to prioritize these values, and attempt to converge towards a final value for the goals. It is also used to monitor the required information, such that graphs and values can be obtained, after the simulation, for these goals.

During the process of simulation, it was also found that simulating the air filter, at certain higher throttle positions, has an impact on the results of the simulation. The filter material is difficult to model as a part by itself, but a solid filter can be modeled, and then set to become a porous material. Running the flow simulation will now incorporate the filter material, as shown in Appendix B on page 81

Improved Model for Environmental Pressure Simulation

During the testing of the effects of an air filter, it was discovered that there was actually a gain in air flow recorded, at low throttle positions, with the addition of an air filter, which seems counter-intuitive as the air filter would seem to act as a restriction for air flow. Stemming from the thought that the air filter had actually assisted the air flow by creating a laminar stream into the throttle body led to the realization that the modeling of components exposed to the environmental atmospheric conditions, should not be modeled with just a “lid” simulating the environmental air, but has to be opened to a significantly larger “chamber” of air.

In the initial preparation for flow simulation, one of the steps was to create a “lid” onto each of the openings of the system. In the improved model, this “lid” is replaced by an octagonal chamber with individual “lids” to simulate the presence of environmental air. The old and new model used for flow simulation are shown below in Figure 24, with the solid blue discs representing the area defined as being at environmental conditions, in this case, since it is the inlet of the air intake system, at atmospheric conditions. The part being modeled and undergoing flow simulation is the throttle body component, with a variable butterfly valve (yellow color component).

What the octagonal chamber represents is a simulation of the infinitely large environment which the throttle body is exposed to in real life. The original lid is itself restricting the air flow into the part shown, and design changes such as a tapering or bell-mouthed lip at the top of the part would not be significant with the original model. On the other hand, the large environment simulated in the new model will allow a difference to be noticed as the design changes.

Figure 24 below shows the two different models.

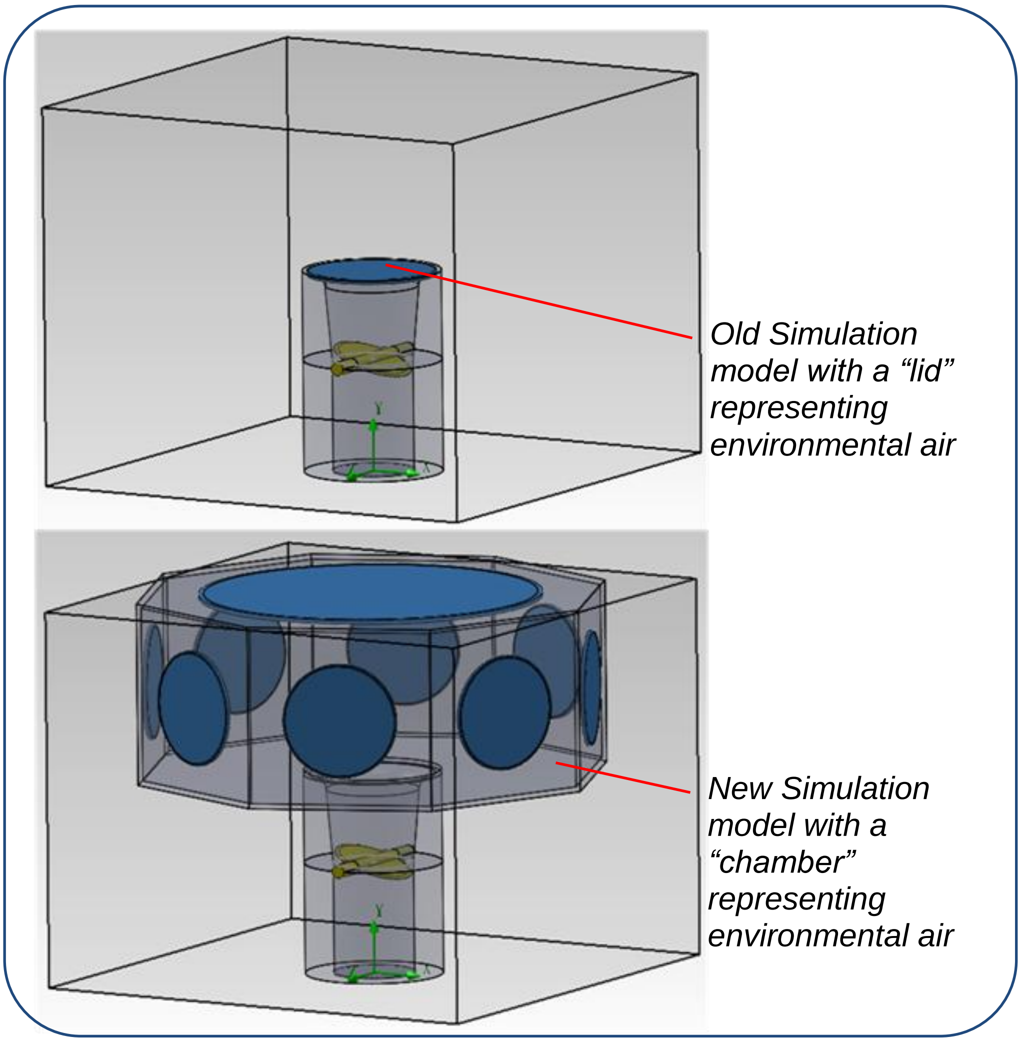


Figure 24: Comparison of Old and New Environmental Air Model

The results highlighting the effect of the change can be found in Appendix B on page 84, where it can be seen that in the old model, there were hardly any noticeable change even though the intake geometry changed from a straight, to a tapered, to a bell-mouth, but on the next page, the changes in airflow, as the geometry changes, is apparent.

FLOWBENCH TESTING

The Flowbench is piece of equipment that is used to test the internal aerodynamic performance of engine components, particularly the air intake system components, such as the cylinder heads and intake manifold. Essentially, the system is made up of a large air pump with an interface to mount the test piece, followed by several chambers for stabilizing air flow and measuring pressure differences to obtain a measurement of the “flow rate” of the test piece.

In the traditional design of the Flowbench, there are a series of chambers through which air will move through, with the flow generated by a suction pump that is placed in-line within, or at the end of, the Flowbench. It will typically make use of measurement devices such as a manometer to measure pressure differences, pitot tubes to measure air speed, as well as an inclined manometer to set the test pressure of the particular experiment. It will often also come with an adjustment valve that will be used to vary the size of a valve to achieve a test pressure (in the metering plenum) such that the reading on the manometer is taken at the same pressure for a particular set of data. A test piece is then placed at the opening of the test plenum, and the necessary parameters, such as valve lift, is varied while the readings are being taken. Figure 25 shows a sample design of a flowbench.

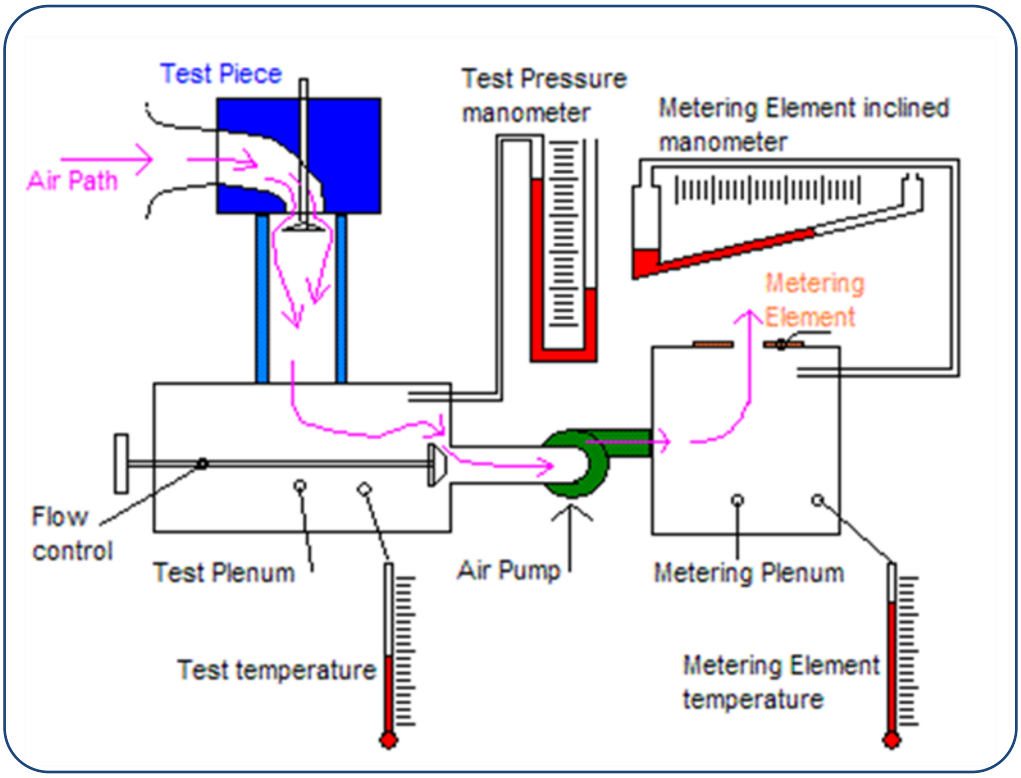


Figure 25: Traditional Flowbench Design

For the Flowbench that is available for use in NUS, it is made with a more “modern” design, using electronic air pressure sensors instead of manometers to derive values such as the flow rate of, and air speed through, an air-flow component such as the intake manifold.

As seen in Figure 26, the Flowbench in NUS is made up of a series of chambers. Between Chamber A and Chamber B, a fixed diameter orifice plate is mounted, with sensors measuring the pressure before and after the orifice plate. Chamber C is used as a pressure stabilizing chamber to allow the air flow to stabilize instead of be affected by environmental transients such as wind. A total of 8 suction pumps are being attached at Chamber C to create the vacuum inside the Flowbench, and thus the airflow through the test piece.

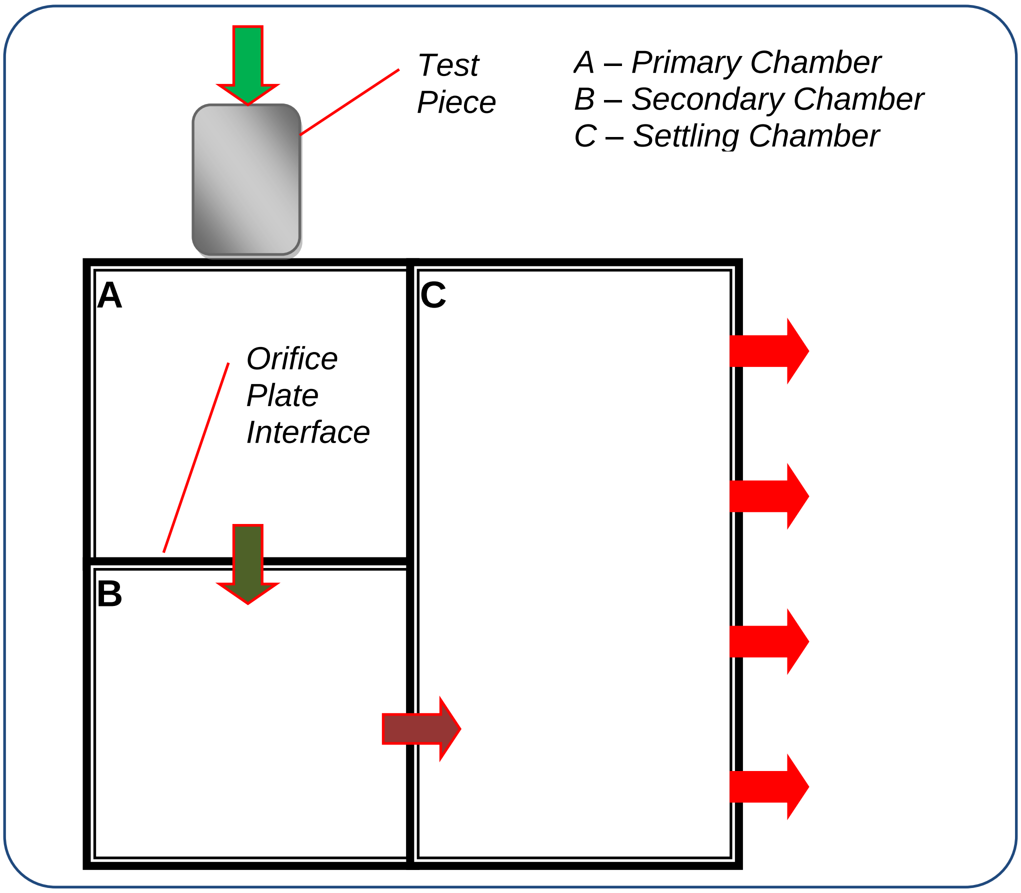


Figure 26: Internal Chambers of Flowbench

Baffle plates in the Primary and Secondary chambers prevent air from flowing directly through the orifice plate or other interfacing holes. This allows the air in each stable to stabilize to a certain pressure before being moved into the next chamber.

Differential pressure measurements between a fixed orifice plate, compared to the differential pressure measurement across the test piece, will give a value for the flow rate through the test piece. The value, however, is insignificant with a mention of the pressure at which the value is obtained. In the traditional system, every test has to be carried out at the same pressure, and this “test pressure” is being achieved by varying a flow control valve. In the system in NUS, the use of electronic sensors has enabled the device to be able to read the test pressure in the Secondary Chamber. Post-processing of this information makes use of the “test pressure” in the Secondary Chamber, to compensate the measured flow rate, and calculate for the projected flow rate at whatever test pressure the data was to be normalized to.

Operation of the Flowbench consist mainly of the preparation work, including designing an interfacing plate for whichever device that is to be measured, so that it can be attached firmly to the Flowbench. The next step would be to mount the test piece onto the interfacing plate, applying gasket to the interfacing plate, before securing it to the Flowbench, allowing 24 hours for the gasket to dry. Figure 27 below shows the required equipment

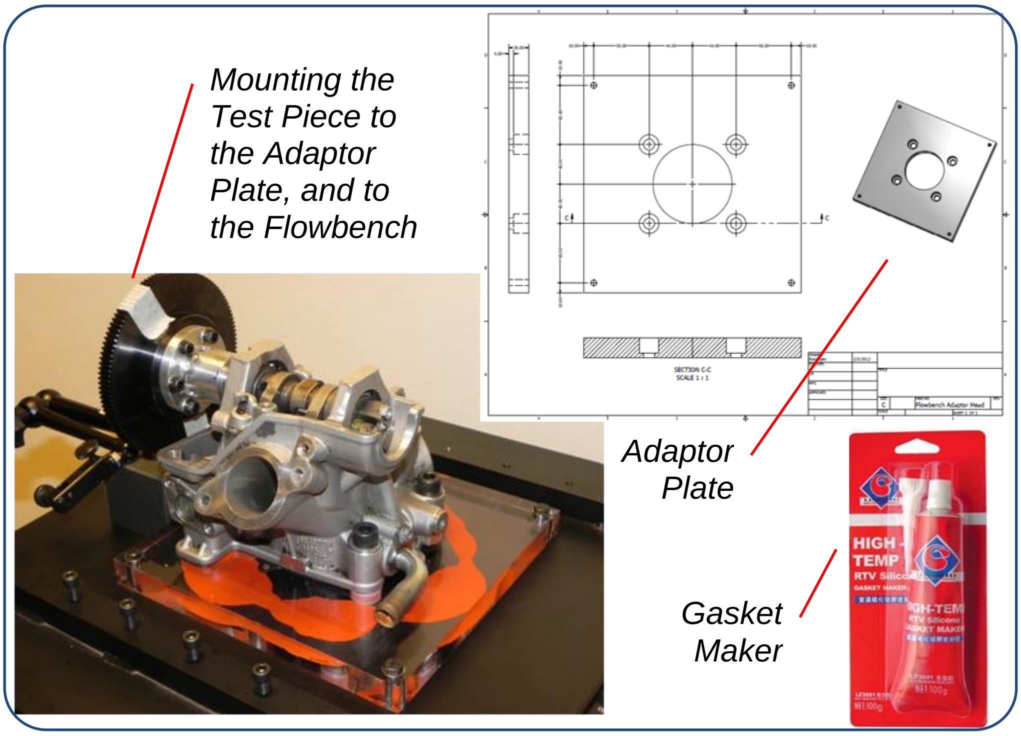


Figure 27: Preparation for using the Flowbench

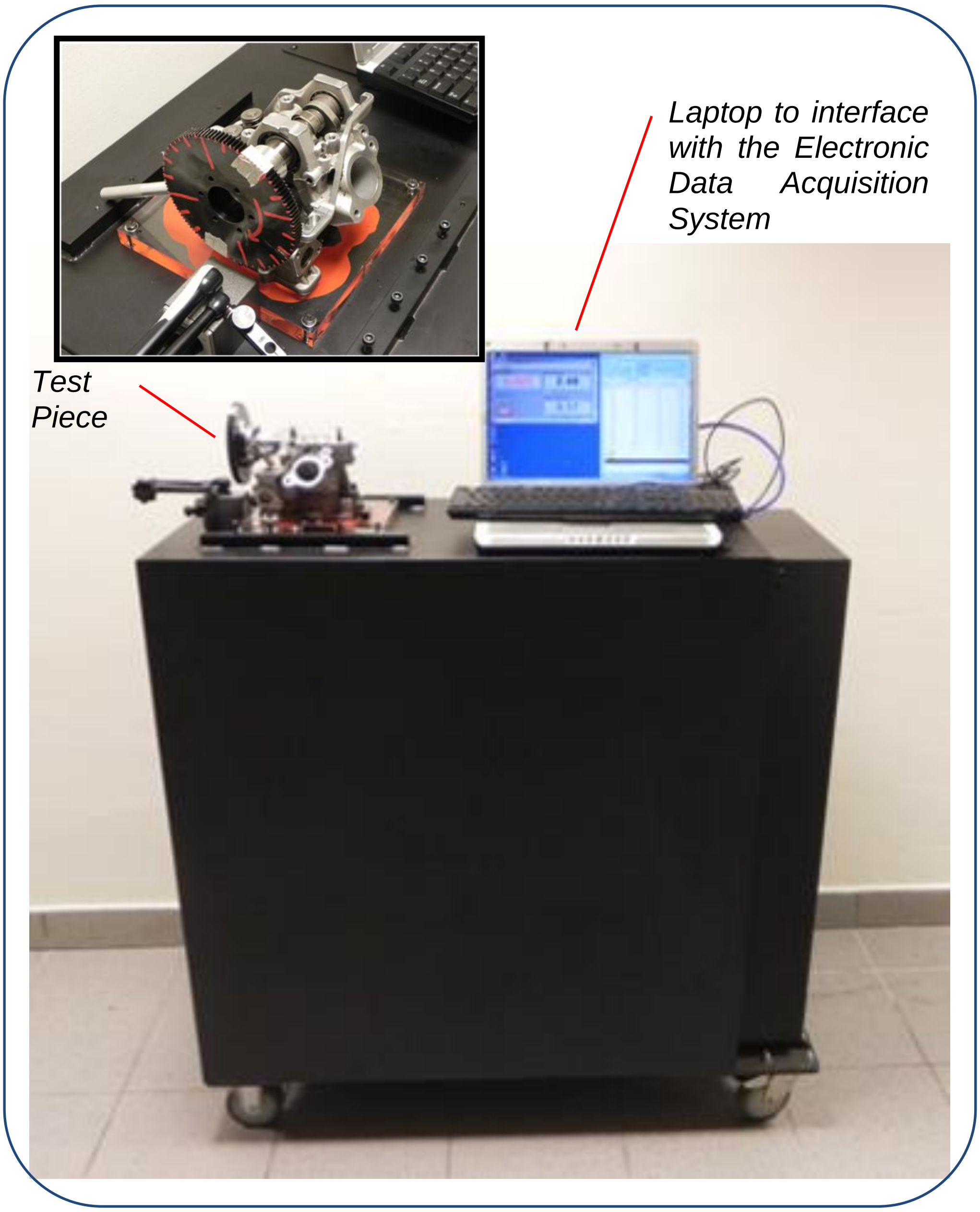


Figure 28: Flowbench in NUS

Figure 28 shows the NUS flowbench with a test piece set up.

Once the test piece has to be mounted, the Flowbench needs to be plugged into a power source, and a "speed" has to be selected. While using single-phase power, the Flowbench can only operate on the two lowest speeds. Further expansion of speeds 3 to 8, will require three-phase power, due to the high loads from the suction pumps. For the context of FSAE intake systems, the first two speeds are sufficient.

Figure 29 shows the Graphical User Interface of the electronic data acquisition system on the flowbench.

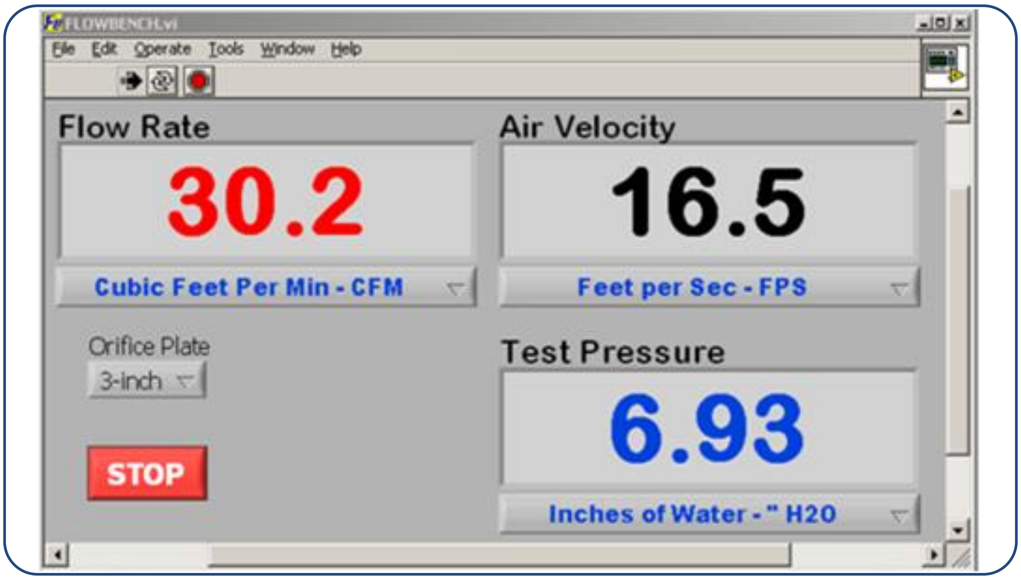


Figure 29: Graphical User Interface of Flowbench

Subsequently, a laptop with the installed Flowbench Data Acquisition User Interface software can be plugged into the USB data interface on the Flowbench. Launching the software produces a panel with readings for Flow Rate, Air Velocity as well as the Test Pressure. Various settings can be done, including adjusting the units of measurements, such as “Cubic Feet Per Minute” or “Cubic Metre Per Minute” for Flow Rates. The test pressure indication shows the vacuum pressure of the Secondary Chamber, shown in “Inches of Water”, a Differential Pressure measurement with respect to the atmospheric pressure. In the screen capture shown above, a 6.93“ H_2O pressure indicates that the vacuum pressure of the Secondary chamber is at a pressure that much below atmospheric. Lastly, an important setting for the Flowbench would be the size of Orifice Plate that is being used, referring to the plate mounted on the interfacing opening between the Primary and the Secondary Chambers.

Design Verification

The Flowbench described in the previous section had been used for an array of data collection for the intake system. One of its first usages was to collect information about the flow rates of the throttle body used on the original Aprilia SXV550 engine.

The throttle body is essentially a pipe with a butterfly valve, through which varying amounts of air can flow through. This is the main form of control that a driver uses to vary the load on the engine, resulting in different speeds of operation. Coupled with a gear box and transmission, this variable operating speed will be translated to the wheels in a compromise of speed and torque.

In order to better understand what the original behavior of the engine was, before the team slaps on an air flow restrictor mandated by the rules, a study on how the original throttle body functions is recorded. This study attempts to co-relate the flow rates through the throttle body and the angle of the butterfly valve inside the throttle body. Initial ideas were to measure the physical angle at which the butterfly valve is at, such as using a dial gauge, but there was no consistent way of doing so without affecting the flow through the test piece.

Therefore, in order to better record the value, a two-stage test was carried out. The first segment of the test was to record the air flow as a function of the Throttle Position Sensor Voltage. The throttle position sensor is essentially a rotary potentiometer which, when given a 5V operating voltage, will produce a range of signals from 0V to 5V, as the butterfly valve rotates over various angles. To record this voltage signal, a microcontroller unit, the Arduino Duemilanove is used. It provides the stable 5V signal required from the sensor, and is able to read in the voltage signal through an Analog-to-Digital Converter on board the microcontroller. The setup is shown in Figure 30 below.

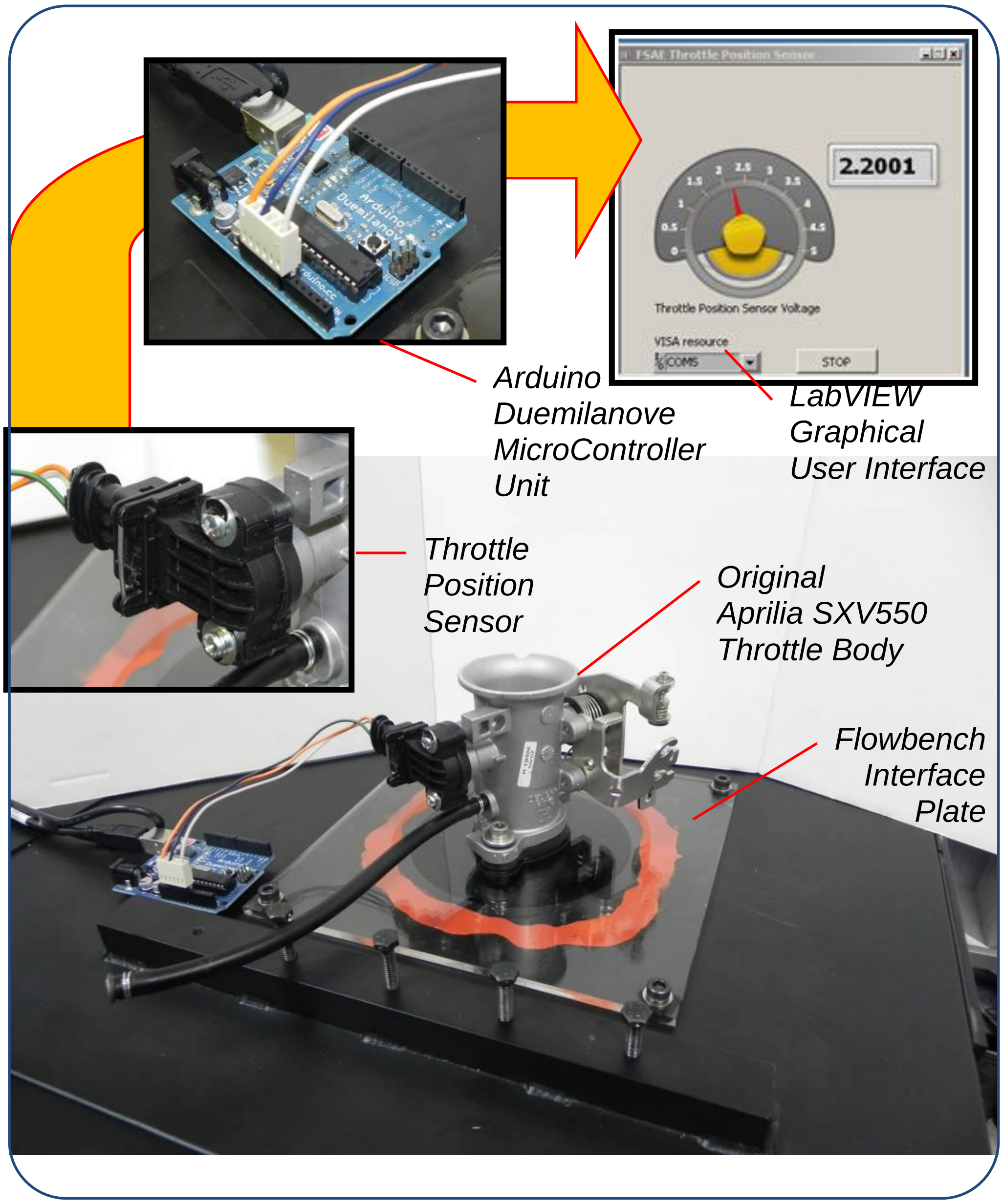


Figure 30: Electronic Setup to Collect TPS Voltage

Testing the throttle requires collecting information about the flow rate at discrete intervals between the “idling” position of the throttle, where it is almost entirely closed (Figure 31 Left) and the “Wide Open Throttle” position (Figure 31 Right) where the butterfly valve is perpendicular to the cross-sectional plane of the throttle body, allowing maximum air flow through. Figure 31 shows the throttle body in the two positions.

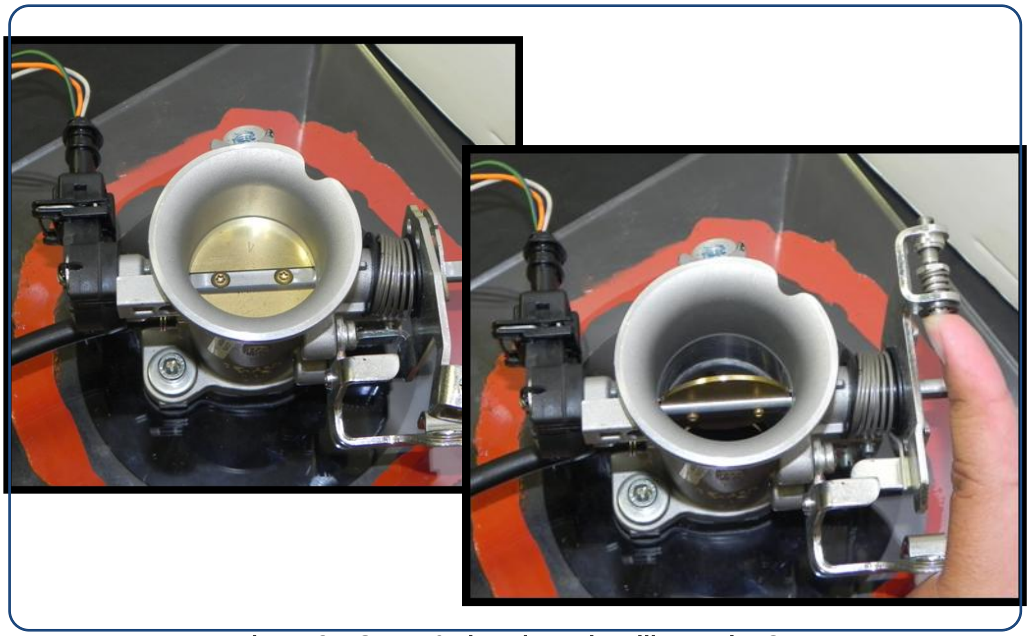


Figure 31: SXV550 Throttle Body, Idling and WOT

As shown in Figure 30, the throttle position sensor is used to measure the angle of the shaft holding the butterfly valve, and the voltage value representing that angle is sent into the Arduino unit. A program had been developed using the LabVIEW software and utilizing the “LabVIEW Interface for Arduino” drivers to develop a Graphical User Interface, quickly displaying the Throttle Position Sensor Voltage value.

In the data collection process, the two User Interfaces for the flowbench, shown in Figure 29 and Figure 30 above can be simultaneously displayed on screen as such:

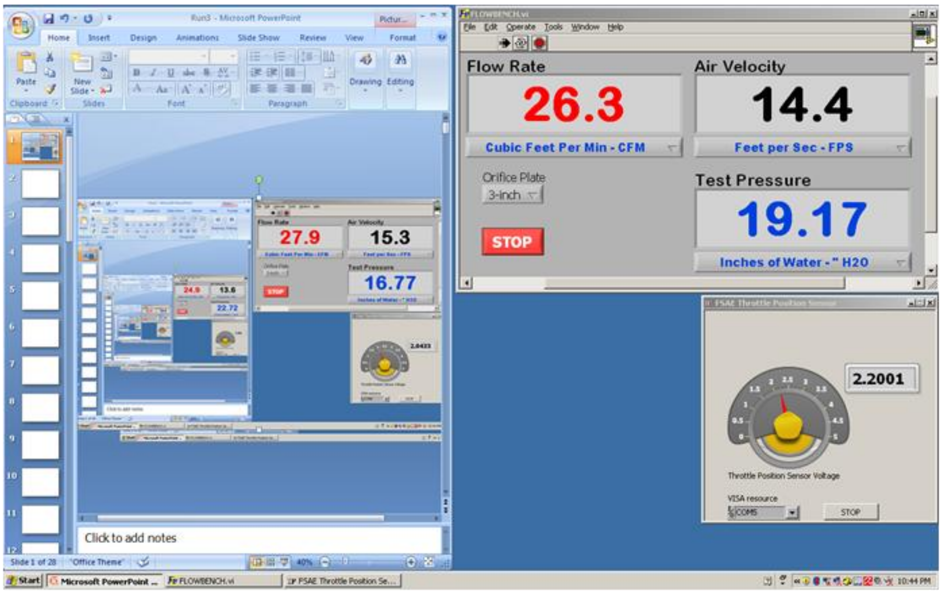


Figure 32: Multiple Graphical User Interfaces for Data Capturing

The screen captures are then collected and tabulated into data files, from which the required relationship of the values can be plotted into graphs. The result of such a test, carried out on the original SXV550 throttle body, carried out over two speeds on the Flowbench, is shown in the graphs in Figure 33 and Figure 34.

As the figures of Flow Rate and Air Velocity had been collected while co-relating only to the voltage reading on the TPS, it is then necessary to co-relate the voltages measured to actual throttle plate angles, which leads to the second stage of the tests. In order to measure the plate angles, a digital inclinometer is being secured to a flat surface on the shaft that holds the throttle plate. As the shaft rotates, the angles will then be co-related back to the voltage values, of which, another graph had been plotted to exhibit the relationship of the two values, as shown in Figure 35.

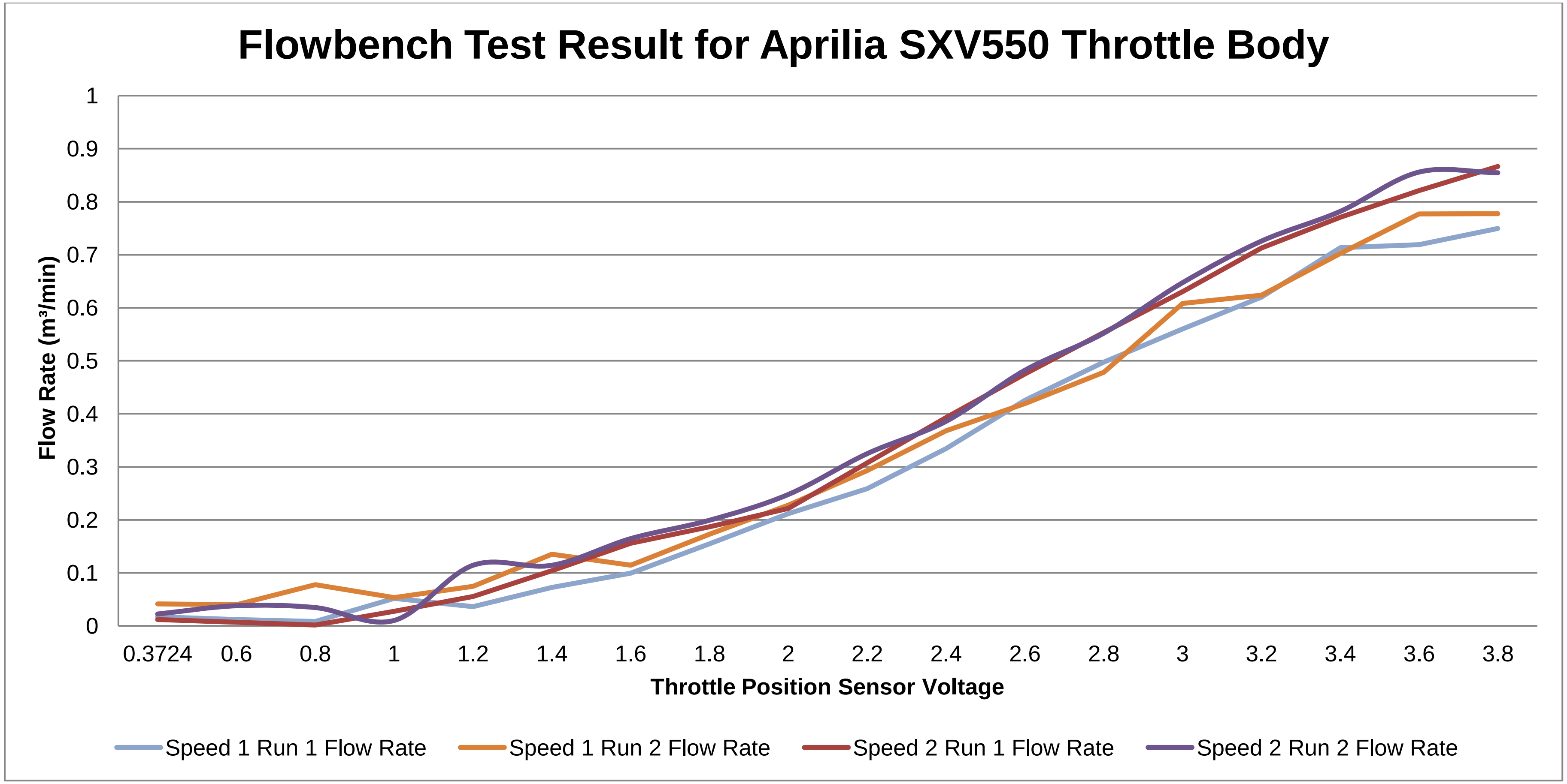


Figure 33: SXV550 Throttle Body Flow Rate vs TPS Voltage

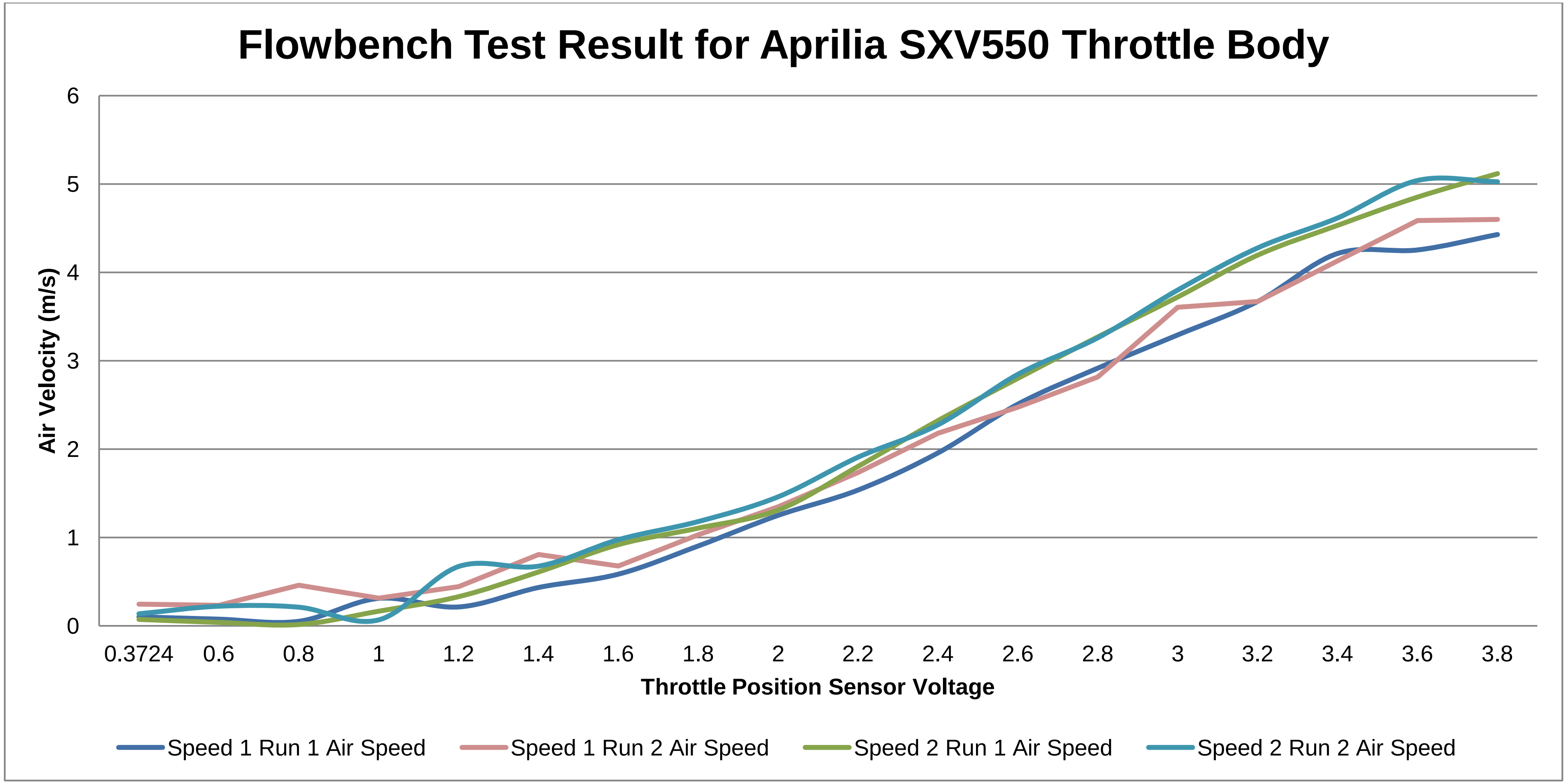


Figure 34: SXV550 Throttle Body Air Velocity vs TPS Voltage

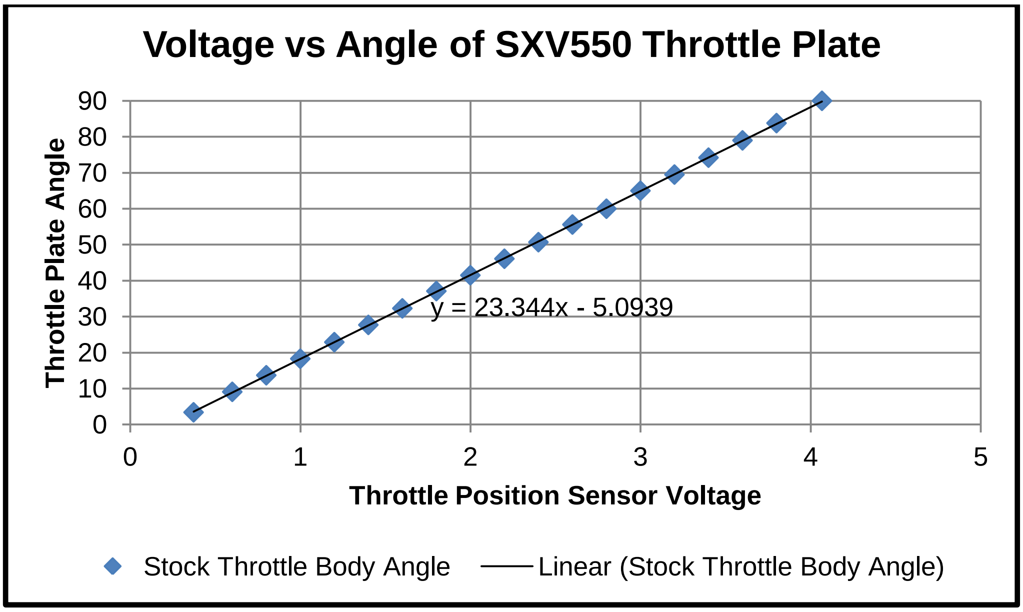


Figure 35: SXV550 Throttle Body TPS Voltage vs Angle

From the graph above, it can be seen that the TPS outputs a Voltage signal at an almost linear relationship with the actual throttle angle, and from the data, a ratio of 23.4° rotation / V of signal change is obtained. As the bespoke throttle bodies for NUS FSAE are also using this sensor, the values can be used to better understand the behavior of the throttle based on the input signal on the TPS.

With the two stages of the experiments completed, a co-relation of the opening of the Throttle Valve Angle and Flow Rate can be achieved. Taking this value a step further, knowing the Throttle Valve Angle will then provide a basis of comparison with a Flow Analysis done on a computer model of the same test piece (graphical results in Appendix B on page 86), collecting data as shown in the following figures:

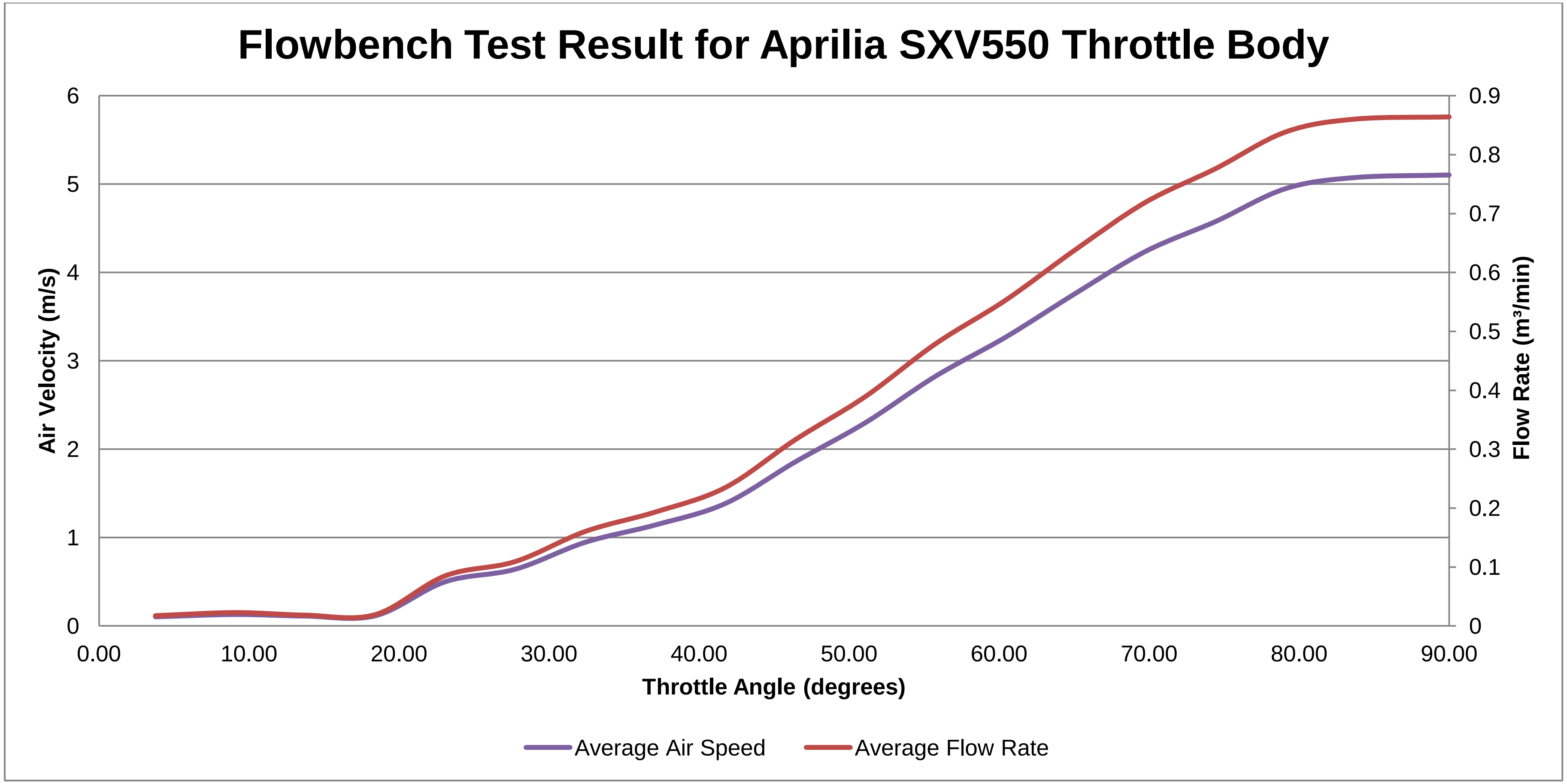


Figure 36: Flowbench Test Results wrt Throttle Angle

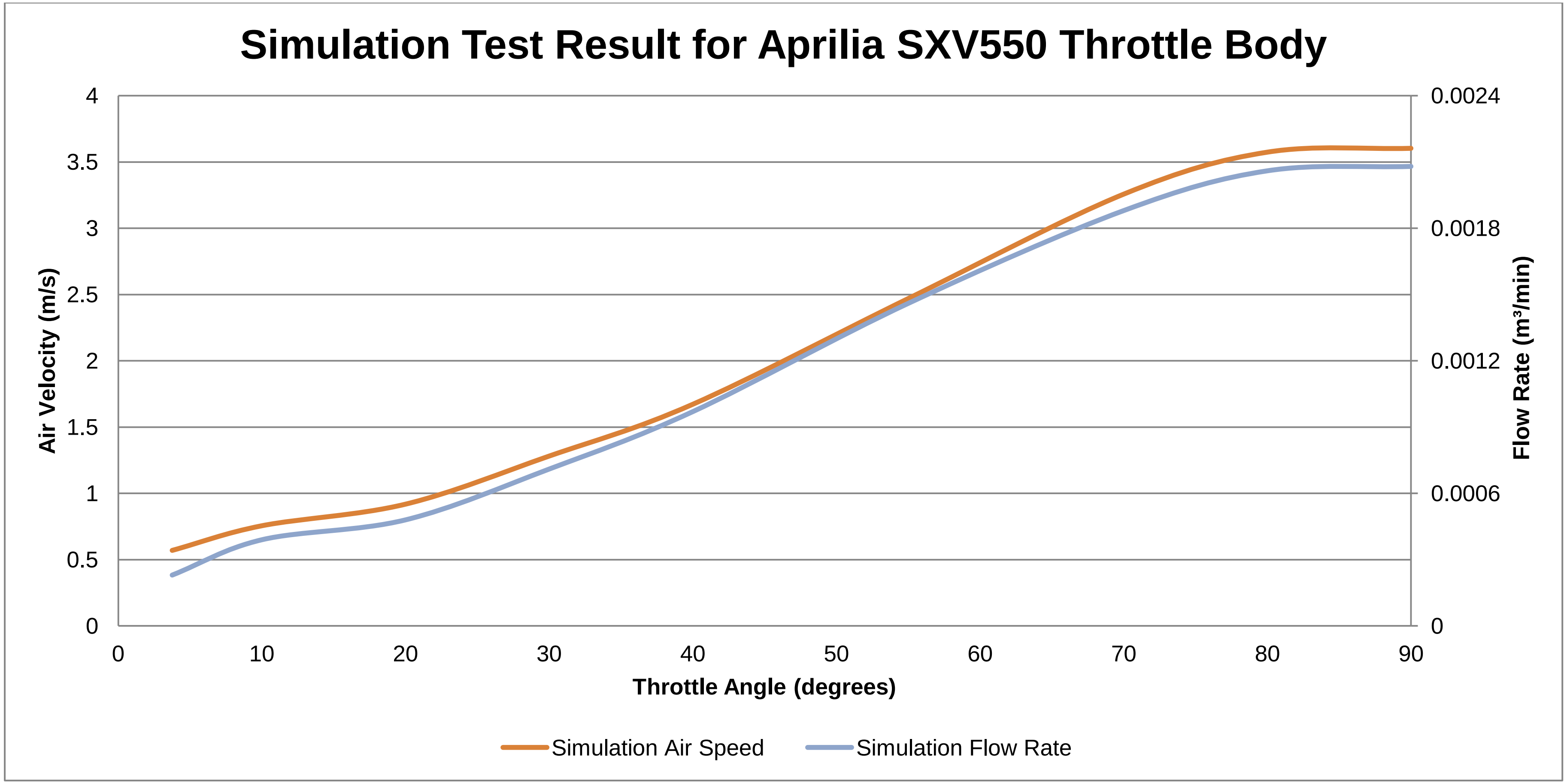


Figure 37: Simulation Results wrt Throttle Angle

5. CONCLUSION

In this report, the design, analysis and testing of a FSAE Air Intake System has been explored. Chapter 2 had given the background of the context of the study, and some prevailing reasons of what the design has to be carried out in a certain way.

The next chapter of the report had dissected the entire system into sub-components, looking into the determining engineering principles behind their designs, and the compromises and trade-offs for certain components.

Finally, the report looks into two essential tools that are used to analyze and verify the design, specifically the Solidworks Flow Simulation Computational Fluid Dynamics software, as well as a Flowbench that uses electronics sensors and a GUI to display the readings.

Overall, it had been a rewarding experience learning about the system and carrying out tests to delve deeper into the engineering behind it and hopefully the report would be a useful reference to shorten the learning curve of the intake system’s design for future NUS FSAE students.

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Coxon, J. (2010, January). A Breath of Fresh Air. *Race Engine Technology.* Issue 043. 66-73.

APPENDIX A ENGINE DATASHEET

ENGINE 550	
ENGINE	
Model (SXV 550)	55SX
Model (RXV 550)	55RX
Type	Twin-cylinder, 4-stroke with 4 valves, single overhead camshaft
Number of cylinders	2
Total displacement	553 cu. cm (33.75 cu in).
Bore/stroke	80 mm x 55.0 mm (3.15 in x 2.16 in)
Compression ratio	(12.5 ÷ 1)± 0.5
Idling speed	(1800 ÷2000) ± 100 rpm
Valve timing (SXV 550) Intake opens at Intake closes at Exhaust opens at Exhaust closes at	 18° BTDC 48° ABDC 49° BBDC 15° ATDC
Valve timing (RXV 550) Intake opens at Intake closes at Exhaust opens at Exhaust closes at	 18° BTDC 30° ABDC 31° BBDC 19° ATDC
Intake valve clearance	0.07 ÷ 0.12 mm (0.0027 ÷ 0.0047 in)
Exhaust valve clearance	0.17 ÷ 0.22 mm (0.0067 ÷ 0.0087 in)
Ignition	Digital, electronic
Starting	Electric starter
Spark advance	Variable, controlled by CDI
Air filter	With dry filter cartridge
Clutch	
Type	Multiplate, wet clutch with control on the left side of the handlebar
Driving plates 1	# plates: 2
	Thickness: 1.5 mm (0.059 in)
Driving plates 2	# plates: 5
	Thickness: 2 mm (0.079 in)
Clutch plates	# plates: 8
	Thickness: 2.75 mm (0.108 in)
Clutch springs	Uncompressed length: 46 mm (1.81 in)
	# springs: 6
Lubricating system	
Type	Gearbox splash lubrication with special fluid; engine forced lubrication with scavenge pump and external reservoir
Oil filter	Paper type
Engine oil quantity	After overhaul, 1400 cu.cm (0.37 gal) (0.31 UKgal)
	Periodic oil change: 1300 cu.cm (0.34 gal) (0.28 UKgal)
	After overhaul, 1400 cu.cm (0.37 gal) (0.31 UKgal)
Cooling system	
Type	Liquid
Water pump	Centrifugal pump with single intake
	Reduction ratio: 44/22

SXV550 TRANSMISSION				
Gear ratios				
Ratio	Primary	Secondary	Final ratio	Total ratio
1st	22/56 = 1: 2.545	12/30 = 1: 2.307	15/48 = 1: 2.875	1 : 16.888
2nd		15/27 = 1: 1.800		1 : 13.172
3rd		16/23 = 1: 1.437		1 : 10.519
4th		20/23 = 1: 1.150		1 : 8.415
5th		21/21 = 1: 1.000		1 : 7.318

FUEL SYSTEM	
Type	Electronic injection
Throttle	Ø 40 mm (1.57 in)
FUEL	
Fuel	Premium-grade unleaded petrol, minimum octane rating 95 (RON) and 85 (MON), as per DIN 51 607.
SPARK PLUGS	
Standard	NGK CR8EB
Spark plug electrode gap	0.7 – 0.8 mm (0.028 – 0.031 in.)
Resistance	5 KΩ
ELECTRIC SYSTEM	
Generator (with permanent magnet)	12 V – 350 W
Starter motor	12 V – 480 W

Figure 38: Specifications from Aprilia SXV550 Technical Manual

APPENDIX B SIMULATION RESULTS

Effect of Air Filter

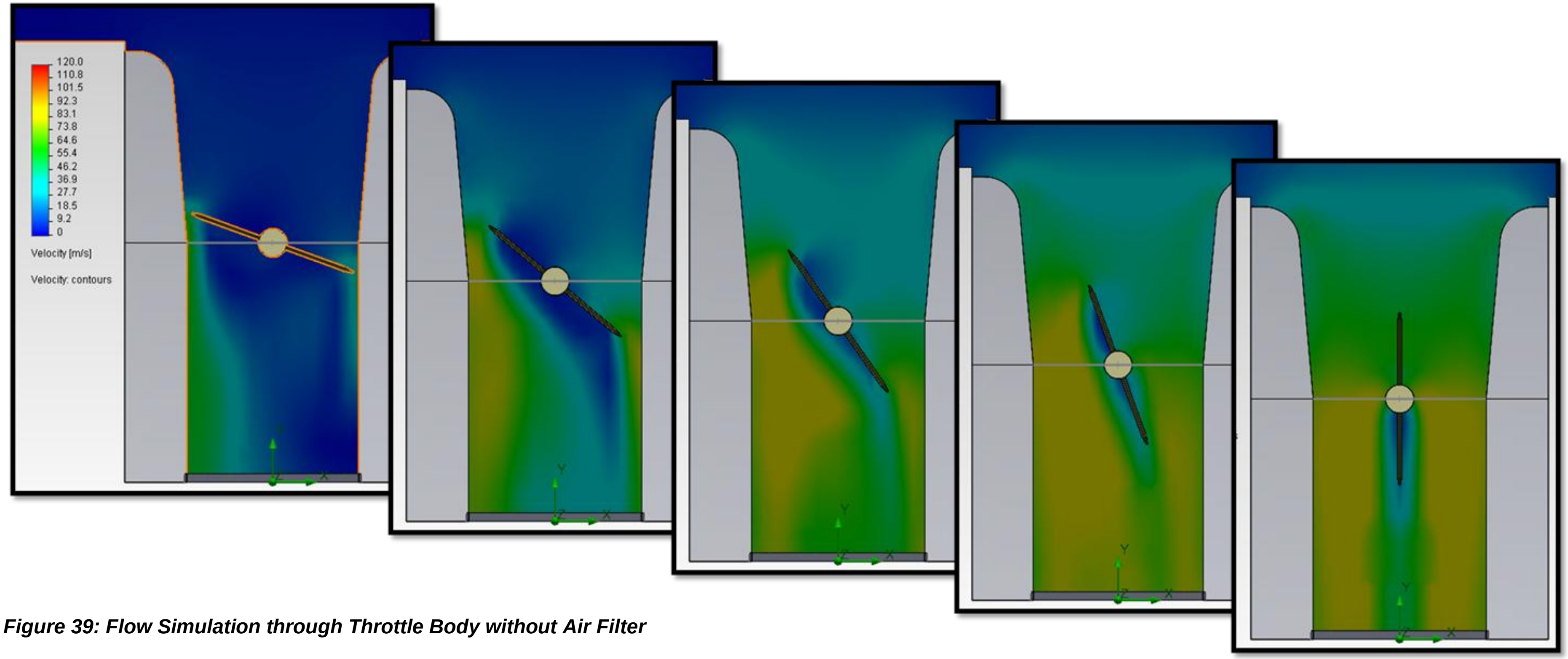

Figure 39: Flow Simulation through Throttle Body without Air Filter

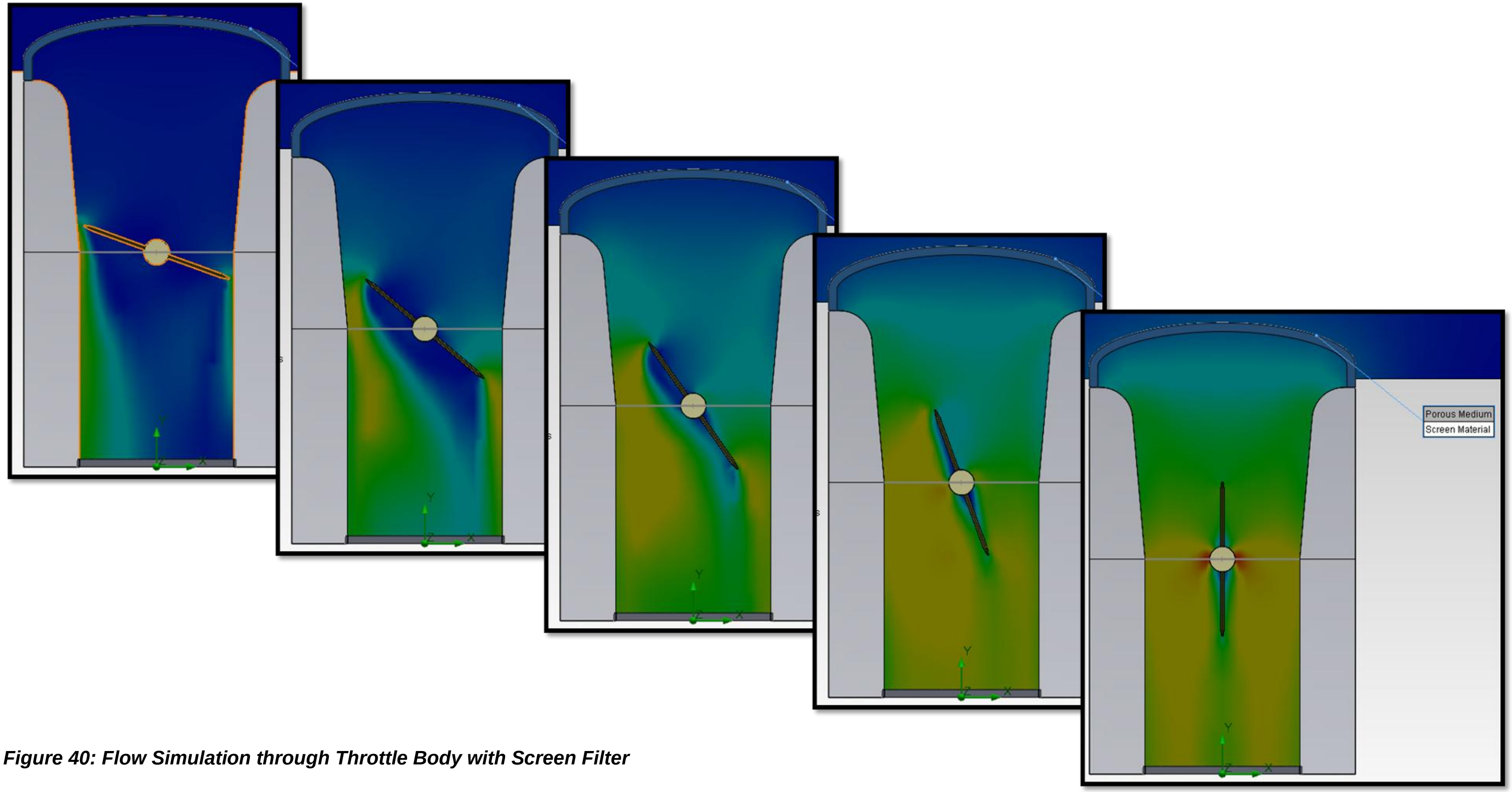

Figure 40: Flow Simulation through Throttle Body with Screen Filter

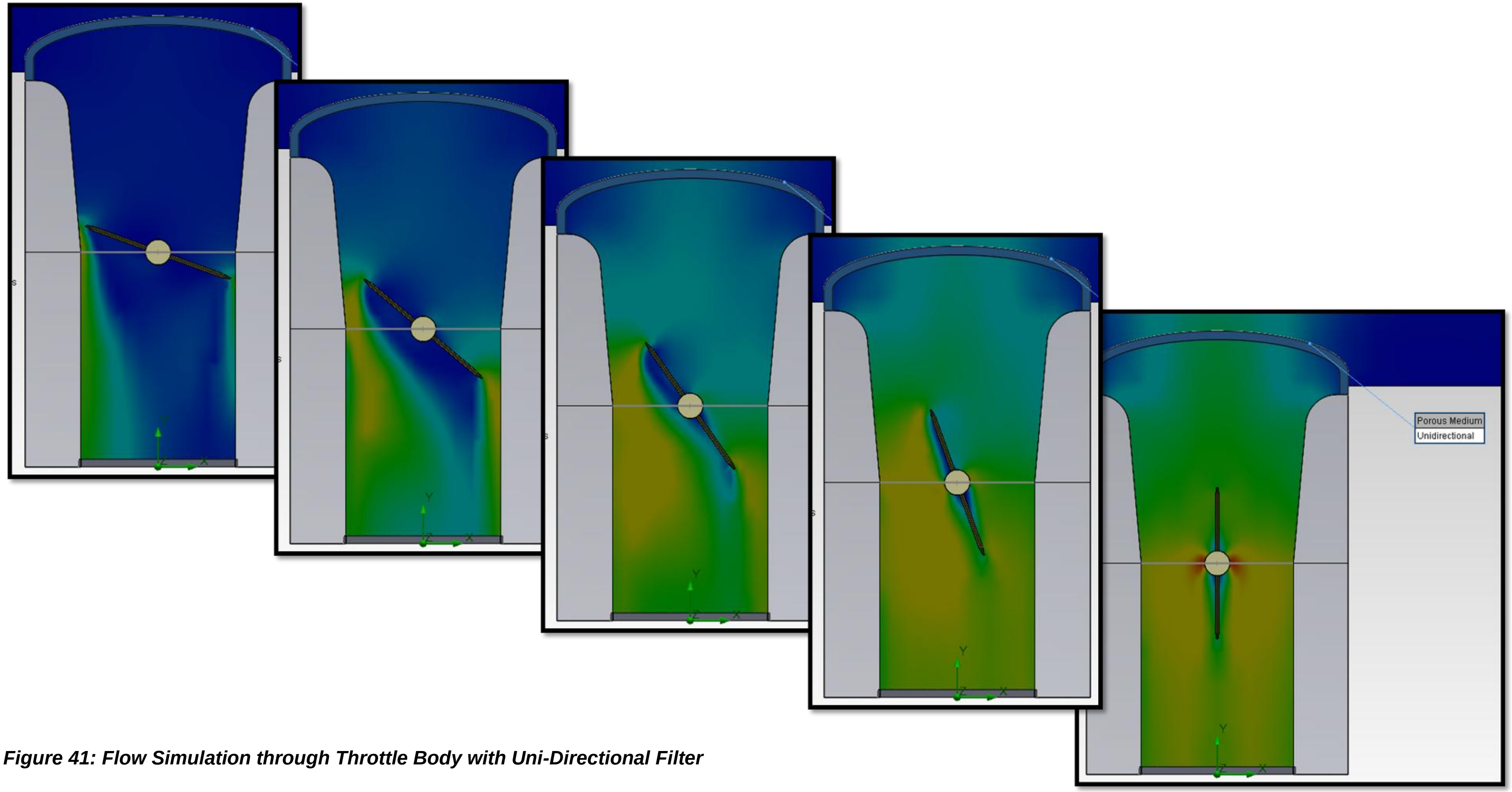

Figure 41: Flow Simulation through Throttle Body with Uni-Directional Filter

Effect of Modified Simulation Model

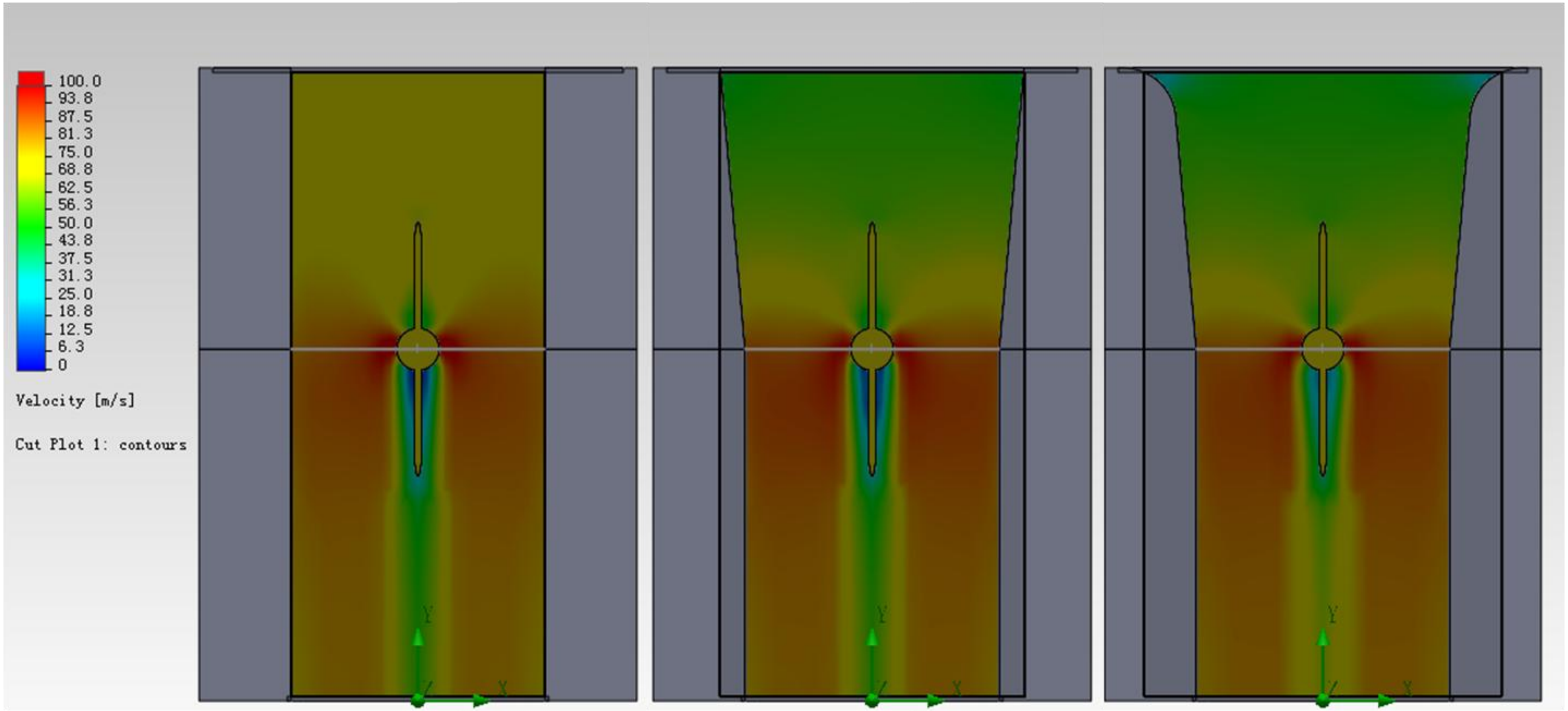

Figure 42: Simulation Results with Original Simulation Model

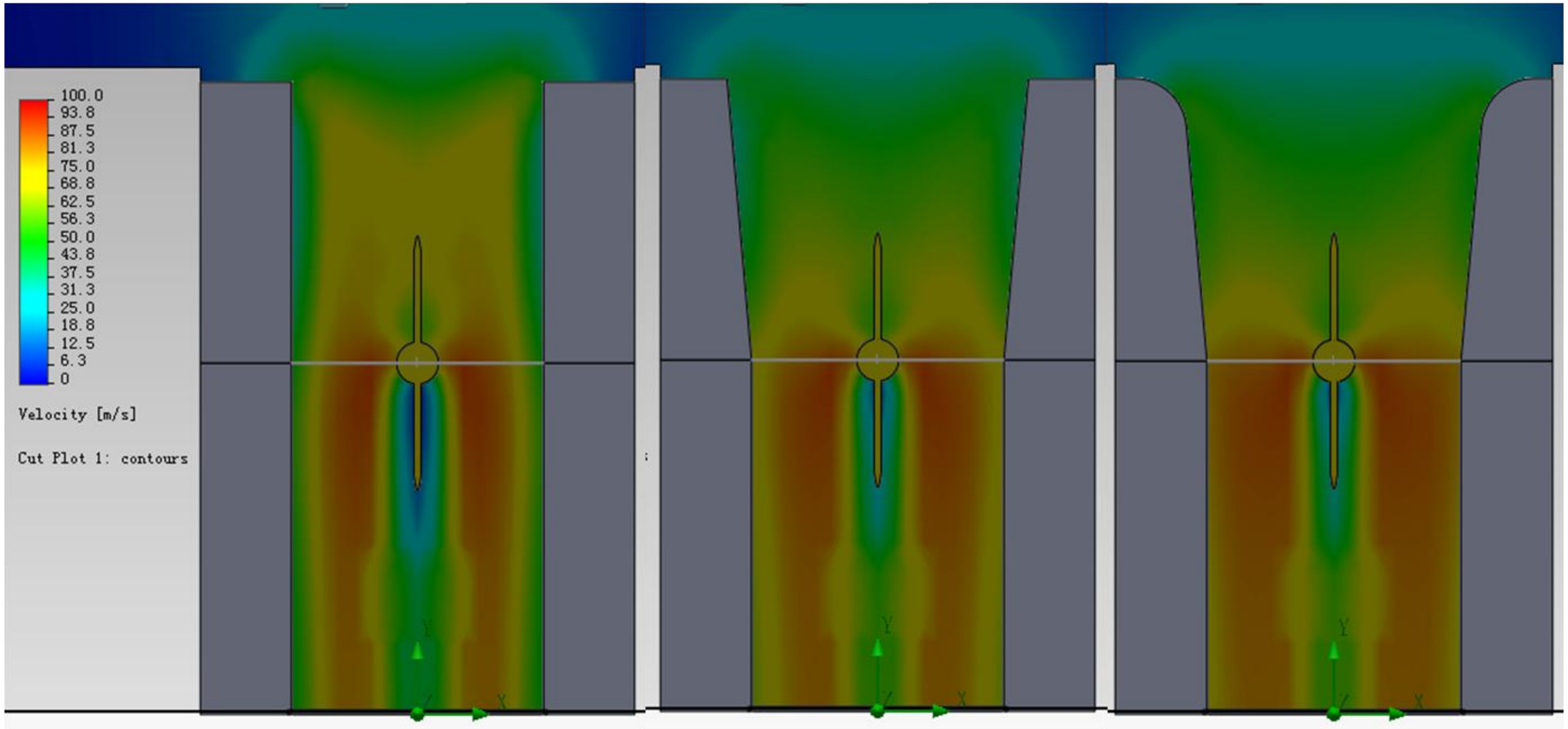

Figure 43: Simulation Results with Modified Simulation Model

Effect of Throttle Angle on Flow

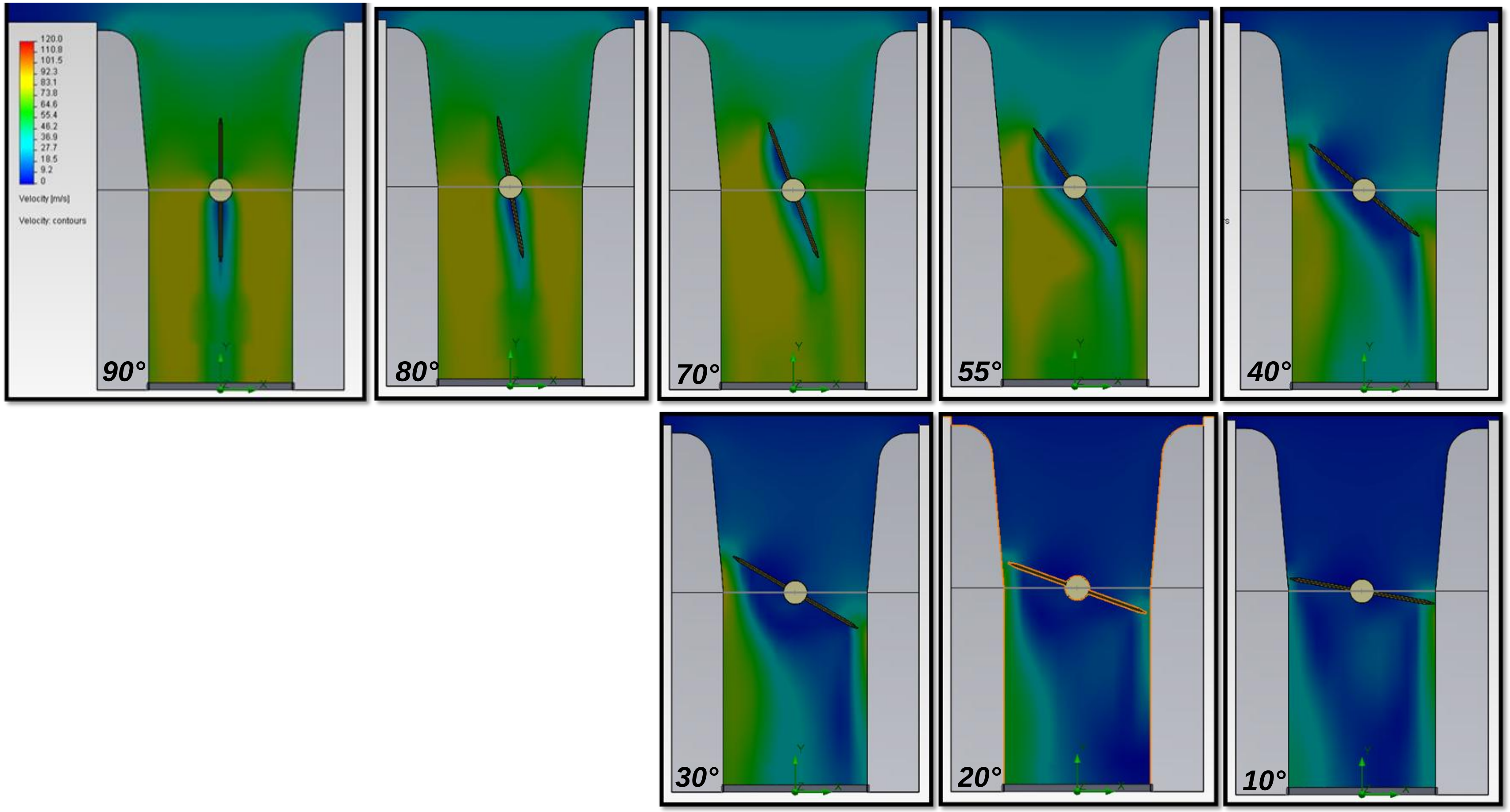

Figure 44: Flow Velocity Pattern for various Throttle Angle Positions